AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

Vol. 43 No. 569 AUGUST, 1953 PRICE: 3s. 6d.

UVA

INTERNAL GRINDING MACHINES

This range of machines is designed for fast and accurate automatic production on long runs. Two methods of automatic sizing are available; solid gauging for through holes, and Deltameter for blind holes or short length bores.

The pressure at which solid gauges are offered to the bore is variable, and the machine auto gauges to limits of 0.0003".

The Deltameter performs an actual check on bore size by the contact of a diamond-tipped pointer within the bore, automatically sizing to limits of 0.0002".

Cross slide feed is continuous, with automatic change-over from rough to finish grinding, and automatic truing of the grinding wheel.

This is the UHM P.195 which is fitted with a Face Grinding Head, shown lowered so that the facing wheel is in the grinding position. Internal grinding capacity ¾" to 8" diameter. There are other U.V.A. Internal Grinding Machines for grinding bores from ½" to 18" dia.

For close tolerance mass production

WICKMAN *of* COVENTRY

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for

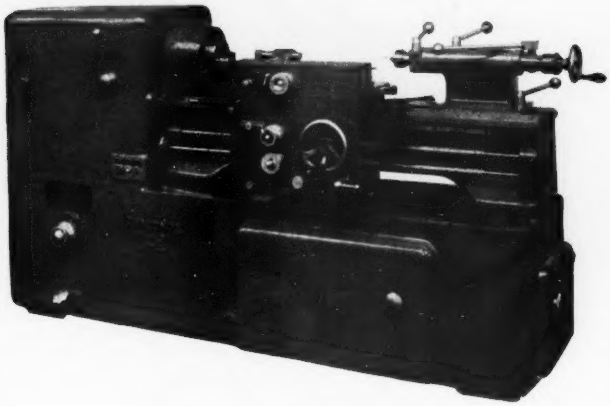
high speed threading . . .

CRI-DAN TYPE 'B' THREADING MACHINE

accurate and highly finished threads produced . . .

USING a single-point tungsten-carbide tool (external or internal, single or multi-start, right-or left-hand, parallel or taper, special or standard), forms can be cut in most materials.

Maximum thread diameter 4″ (external), 6″ (internal); 3′ 0″ admitted between centres.



Typical production obtained is shown at:—

A. Focusing mount, aluminium. Spiral groove ·100″ wide × ·050″ deep, ½″ lead, 2″ long on 1⅞″ diameter. Cutting time 11 seconds.

B. Diesel Stud, 70 tons tensile steel. ⅞″ × 11 t.p.i. Whit. 1¾″ long. Cutting time 17 seconds.

C. Adapter, mild steel. 1⅜″ × ⅜″ long (internal) 1⅝″ × ¼″ long (external), 14 t.p.i. Whit. Cutting time 8 seconds (two threads cut simultaneously).

D. Hose coupling, phosphor bronze. 2¾″ × 2¼″ long, 8 t.p.i. Whit. and 3⅛″ × ·280″ deep × 1⅜″ long, 2 t.p.i. Knuckle. Cutting time 25 and 40 seconds Whit. and Knuckle respectively.

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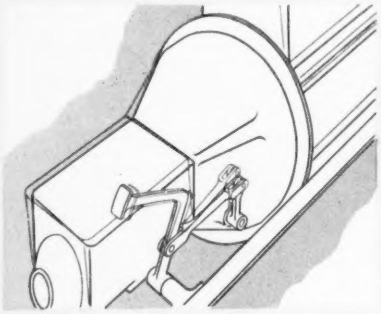
TRANSMISSION CONSIDERATIONS 3

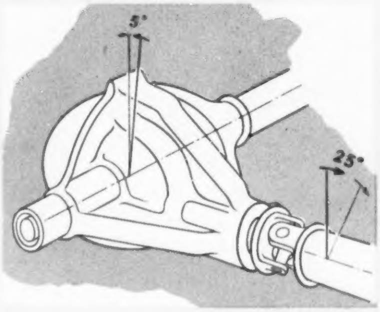
JUDDER (2)

Two of the simpler causes of judder were discussed in a recent article. Both were removable without great difficulty. A faulty sparking plug, indeed, is scarcely a serious hazard today; and, as was indicated, there are several possible methods of 'liquidating' the unpleasant effects of longitudinal motion of the power unit in its flexible mountings.

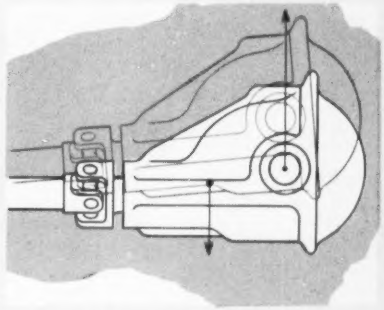
However, flexible mountings can cause judder also by permitting the power unit to oscillate about a longitudinal axis. There is at least one recorded case in which the judder condition was reversed by conversion of the design to left-hand drive. Clearly the connecting end of the clutch-shaft drop-arm would rise if it were on the one side whereas it would fall if on the other, the engine motion being the same in both cases.



Interference of clutch linkage by rocking motion of engine.



Axle-rotation due to torque reaction.



Every bump tends to rotate a Hotchkiss rear axle.

Appreciation of the circumstances led to a scheme midway between these two, so that this source of judder was eliminated whether the pedal was on the right or on the left. If no such simple solution were possible, the trouble could be removed by using a servo mechanism or by hydraulic operation of the clutch.

It would be unfortunate to give the impression that judder always originates in or about the power unit. Experience suggests that it can start almost anywhere in the vehicle. There is for instance a commercial chassis which judders when carrying a tipping body but is free from the fault when bearing a coach body. Presumably the difference is one of overall stiffness of the complete vehicle.

Incidentally, the student of design could spend a little time in many worse ways than by watching heavy vehicles approaching and leaving a loading bay. Usually they are high enough from the ground to display fairly clearly some at least of the peculiar effects of heavy and intermittent torque. A natural extension of this little lesson would be to continue the observation while following the lorry in a reasonably low car.

Still another cause of judder is oscillation of the axle casing on its springs — quite apart from the effect, mentioned earlier, that this may have in moving the power unit longitudinally.

Suppose that the vehicle is at rest with its driving wheels on the ground. Suppose further that the axle casing is turned through a small angle by any suitable means. This will carry the bevel pinion around the crown wheel and, the latter being held by the axle shafts, the bevel pinion must rotate on its own axis. Moreover, a little consideration will show that, owing to the gear ratio between pinion and crown wheel, the former will rotate several times as much as the axle casing.

By way of example, if the axle ratio is 5 to 1 and the casing is rotated through 5 degs. then the pinion will turn through 25 degs. This motion will be transmitted to the propeller shaft, of course, and so forward through the rest of the transmission system.

It should not be overlooked that a precisely similar effect occurs also when the vehicle is in motion. Any angular motion due to the partial rotation of the axle casing is then impressed upon the normal movement of the various parts. Moreover, such rotation occurs with every appreciable change of throttle because this alters the torque applied to the casing. Since the latter is carried by springs which, in this respect, are virtually undamped, displacement in one direction will be followed by rebound which will overshoot and so on. In short, the casing will oscillate and thus repeated reversals of motion will be applied to the transmission.

If the latter is moving only slowly, as when starting from rest, these applied reversals may over-ride the steady speed caused by the engine. In such a case there will be judder which, as we noted earlier, is outward evidence of reversals of load at the various clearances in the transmission.

Axle oscillation can occur apart from changes of throttle. Clutch engagement alters the torque applied to the axle. If that engagement is rapid, the torque change may be sufficiently sudden to cause oscillation.

There is also the fact that the axle casing is not in rotational balance. Its nose-piece and the propeller shaft have no counterweight. Violent displacement of the axle by road irregularities must therefore tend to turn the casing because the force acting at the wheel centre does not pass through the c.g. of the displaced mass.

Except in this last instance, shock absorbers cannot help very much. Axle oscillation is in fact an inherent fault of Hotchkiss drive. It is part of the price to be paid for such attractions as this system offers in other directions.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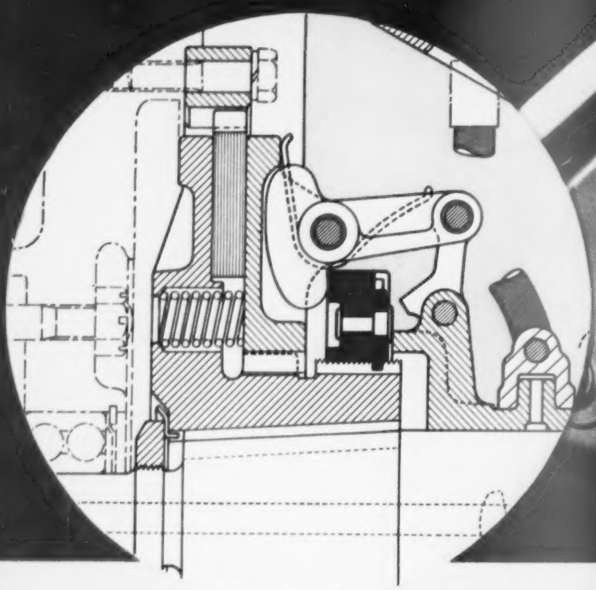

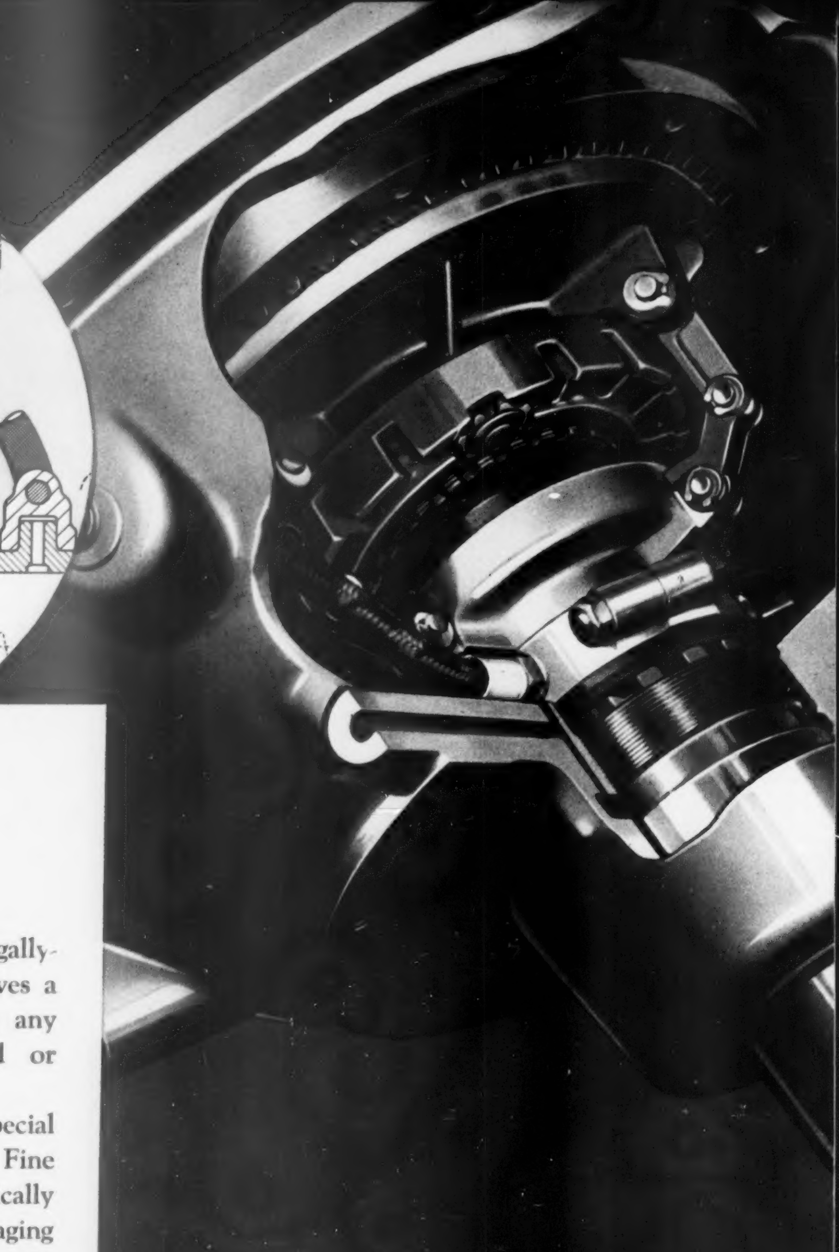

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An exclusive device gives, without special tools, an infinitely close adjustment. Fine adjustments can be made, and automatically maintained, without releasing or engaging separate locking devices, which formerly limited adjustments to spacing controlled by notches or holes. The adjustment ring is conveniently accessible, and is painted red for easy identification.

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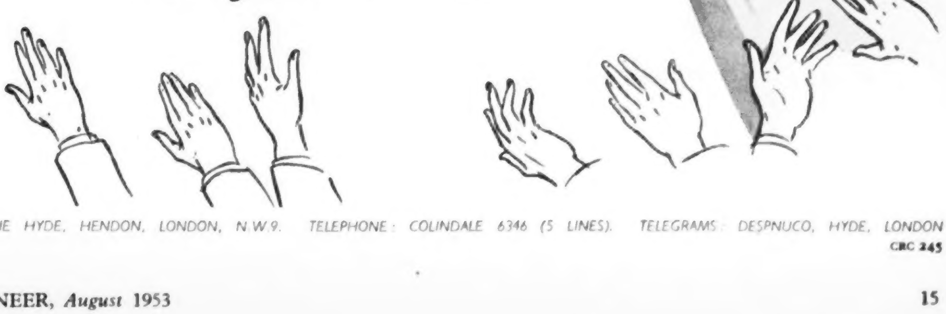
From almost every country in the world the demand for Desoutter Power Tools increases month by month. Only by exerting yourselves to the . . .

(But at this moment the hooter announced the end of the luncheon break and the eager workers broke away to their benches hungry but happy)

Desoutter tools *put power into your hands*



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CRC 245

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These tools are now being made in Wimet tungsten carbide with precision ground teeth, in a range of six standard shapes and the most commonly used sizes.

You can order them from any Wickman Area Office, from stock. Simply quote the reference number, quantity required, and state whether fine or coarse cut.

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All Wimet Burs listed overleaf are available with either fine cut for slow speeds or coarse cut for high speeds.

P.T.O. for sizes and prices

IN STOCK at all Wickman Area Offices

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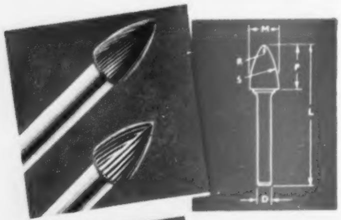
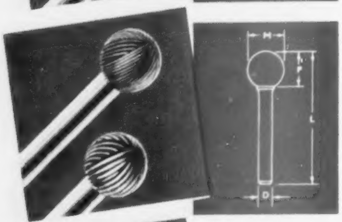


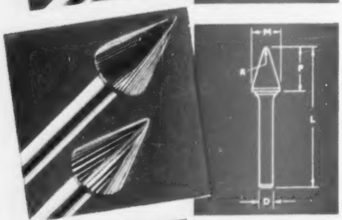
TYPE P

REF.	M	L	D	P	R	S	PRICE
P1	1/4"	2 1/8"	·2355"	1/2"	1/32"	1 1/4"	£2 15 0
P2	3/8"	2 1/4"	·2355"	5/8"	1/32"	1 1/4"	£4 2 6
P3	1/2"	2 3/8"	·2355"	3/4"	1/32"	1 5/16"	£5 10 0



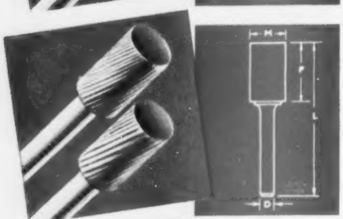
TYPE S

REF.	M	L	D	P	PRICE
S1	1/4"	2"	·2355"	7/32"	£2 10 0
S2	3/8"	2"	·2355"	11/32"	£3 10 0
S3	1/2"	2 1/16"	·2355"	·460"	£4 15 0
S4	5/8"	2 1/4"	·2355"	·583"	£6 10 0



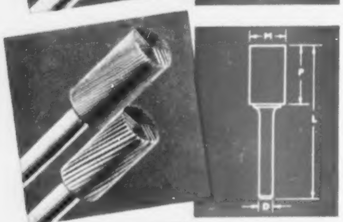
TYPE C

REF.	M	L	D	P	R	PRICE
C1	1/4"	2"	·2355"	1/2"	1/32"	£2 15 0
C2	3/8"	2 1/4"	·2355"	5/8"	1/32"	£4 2 6
C3	1/2"	2 3/8"	·2355"	3/4"	3/64"	£5 10 0



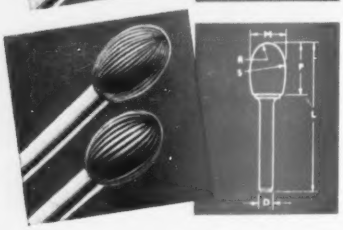
TYPE F

REF.	M	L	D	P	PRICE
F1	1/4"	2"	·2355"	3/4"	£2 17 6
F2	3/8"	2 1/2"	·2355"	7/8"	£4 0 0
F3	1/2"	2 5/8"	·2355"	1"	£5 5 0
F4	5/8"	2 5/8"	·2355"	1"	£6 15 0



TYPE N

REF.	M	L	D	P	PRICE
N1	1/4"	2"	·2355"	3/4"	£3 2 6
N2	3/8"	2 1/2"	·2355"	7/8"	£4 7 6
N3	1/2"	2 5/8"	·2355"	1"	£5 15 0
N4	5/8"	2 5/8"	·2355"	1"	£7 5 0



TYPE B

REF.	M	L	D	P	R	S	PRICE
B1	3/8"	2 1/4"	·2355"	5/8"	3/16"	2 1/2"	£4 2 6
B2	1/2"	2 1/2"	·2355"	7/8"	1/4"	3"	£5 10 0
B3	5/8"	2 1/2"	·2355"	7/8"	17/64"	1 1/8"	£7 0 0

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CARBO-NITRIDING IN BIRLEC-DOW FURNACES

Carbo - nitriding replaces liquid cyaniding by a gaseous process, giving equivalent cases at much lower operating costs. With a greater penetration rate, lower temperatures can be used, thus reducing distortion.

The BIRLEC-DOW furnace

★ A gas-fired batch unit, enabling a wide range of components to be heat-treated.

★ Totally enclosed, heating and quenching being carried out without exposure to the air.

★ Built-in atmosphere generator, cutting down floor space requirements.

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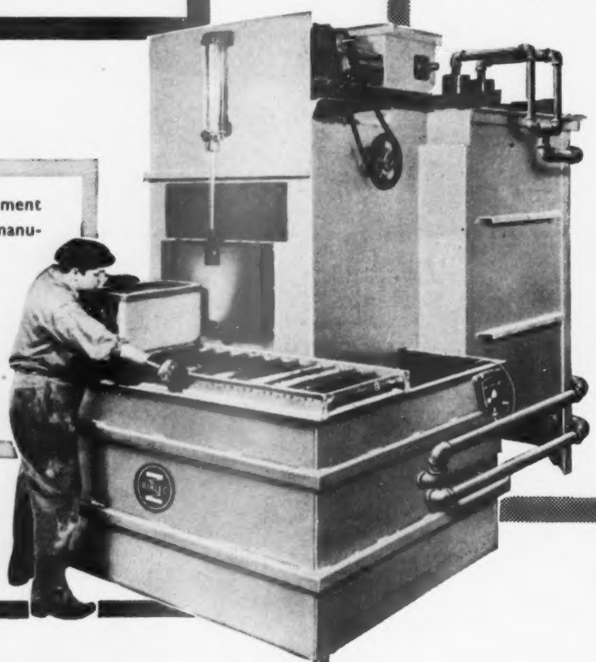
★ Efficient atmosphere circulation, permitting dense loads of closely packed articles to be handled.

★ Semi-mechanised, reducing labour requirements and providing good working conditions.

By exclusive arrangement with the American manufacturing patentees, BIRLEC LIMITED have now added the DOW carbo-nitriding furnace to their wide range of heat-treatment equipment.



Standard designs are available from 250 lb. per hour (carbo-nitriding with .003" case) upwards. Full design and operating data are available; **may we send further details?**

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"Elektron"
LIGHTER CASTINGS
FOR HEAVY VEHICLES
WHICH ARE 40% LIGHTER THAN
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Detachable oil filter

element easily inspected and replaced

This new AC partial-flow oil filter with replaceable element meets the increasing demand for a filter that can be removed for inspection and replaced every 8/10,000 miles. It provides a permanent installation for coupling into a by-pass oil supply system and the filtering element can be replaced without disturbing pipe connections. Sump oil is filtered on an average of ten times an hour. This means less motor wear and longer motor life — the objective of every motor manufacturer.

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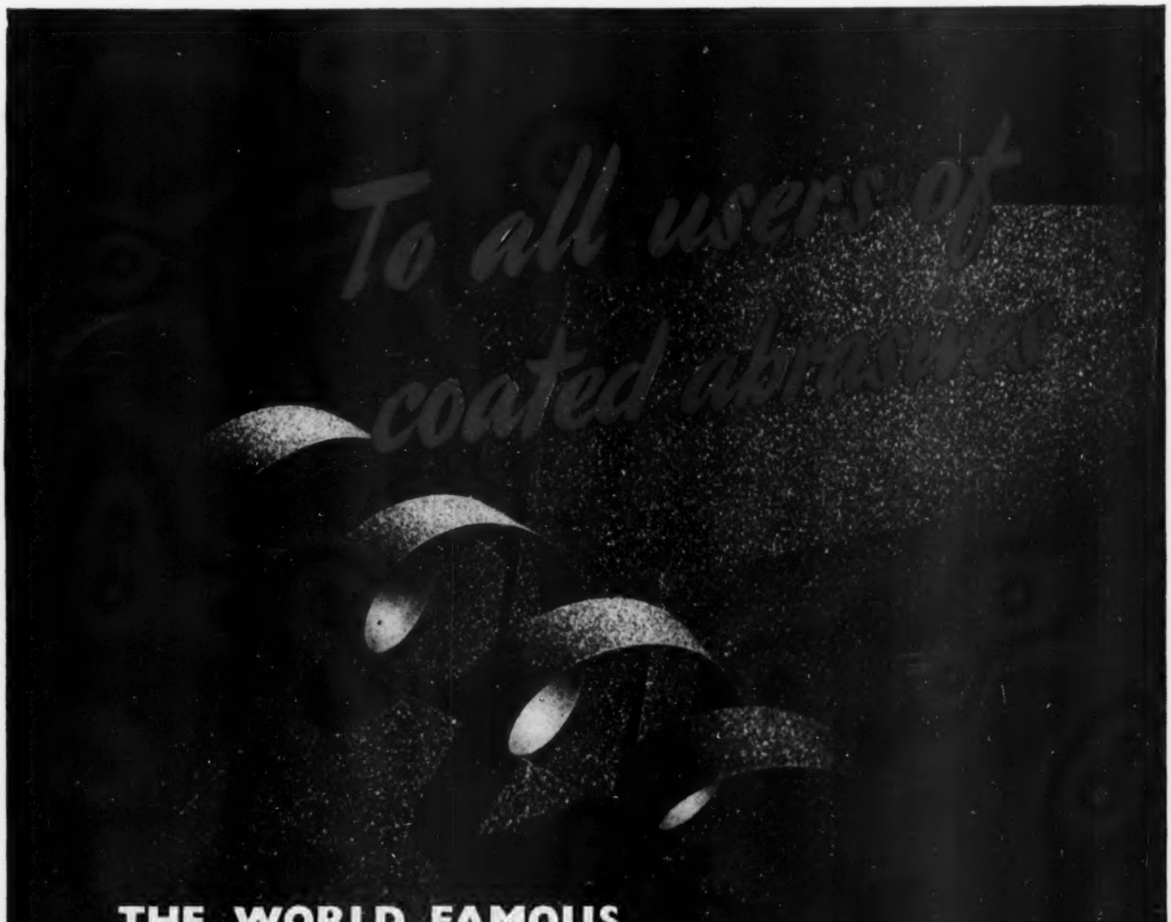
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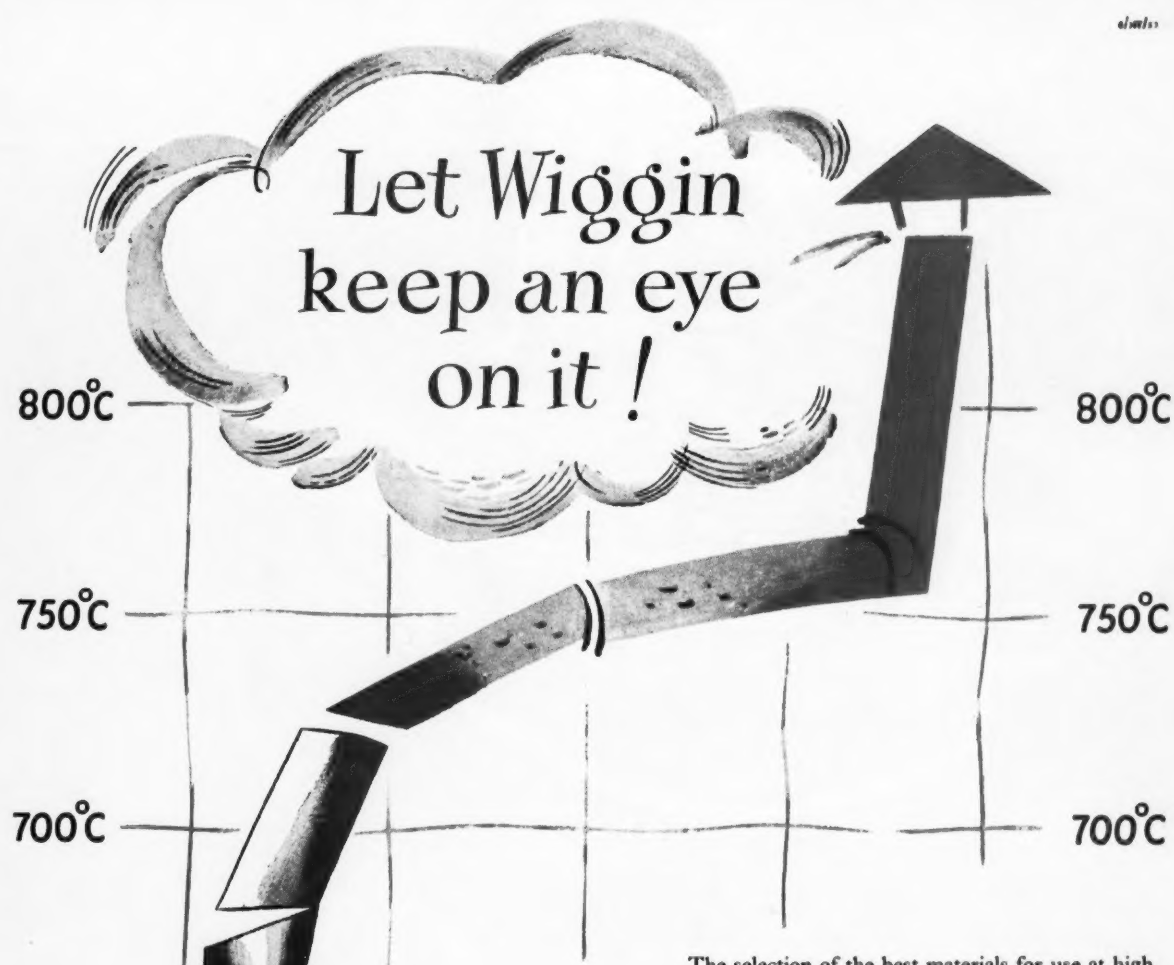

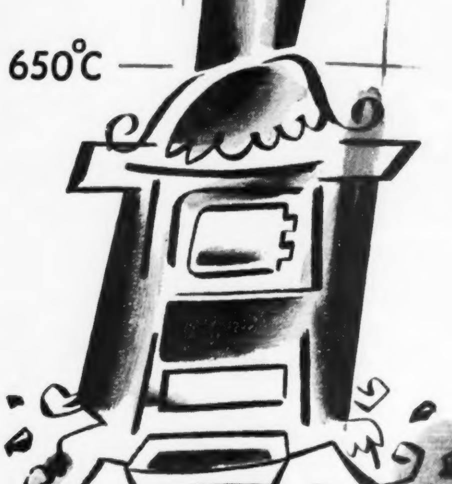

The selection of the best materials for use at high temperatures is a problem for the expert. That is where we can help. We have research data and service results covering the use of a wide range of materials in various operating conditions. Amongst this wealth of technical information we may already have the solution to your own high-temperature problem. The chances are that one of our high-nickel alloys will best meet your requirements — but ask us *before* you go ahead.

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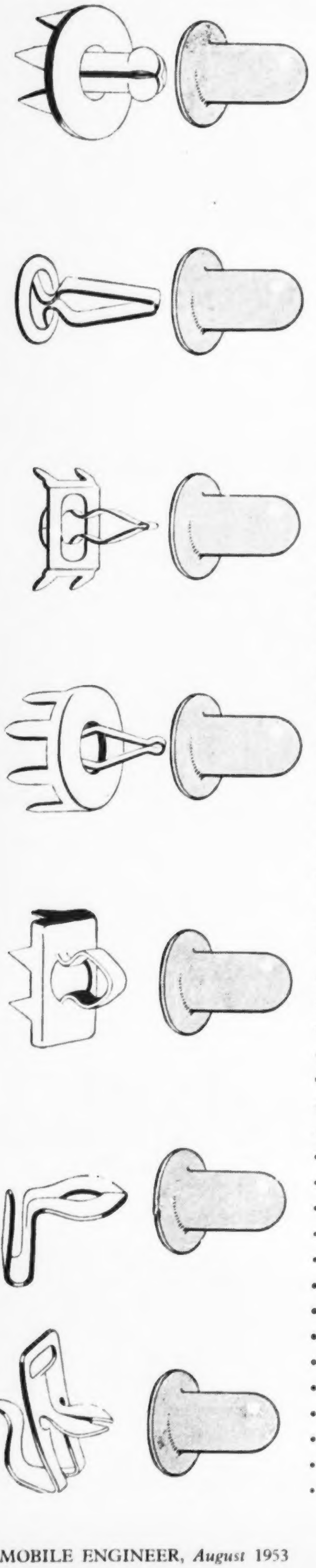
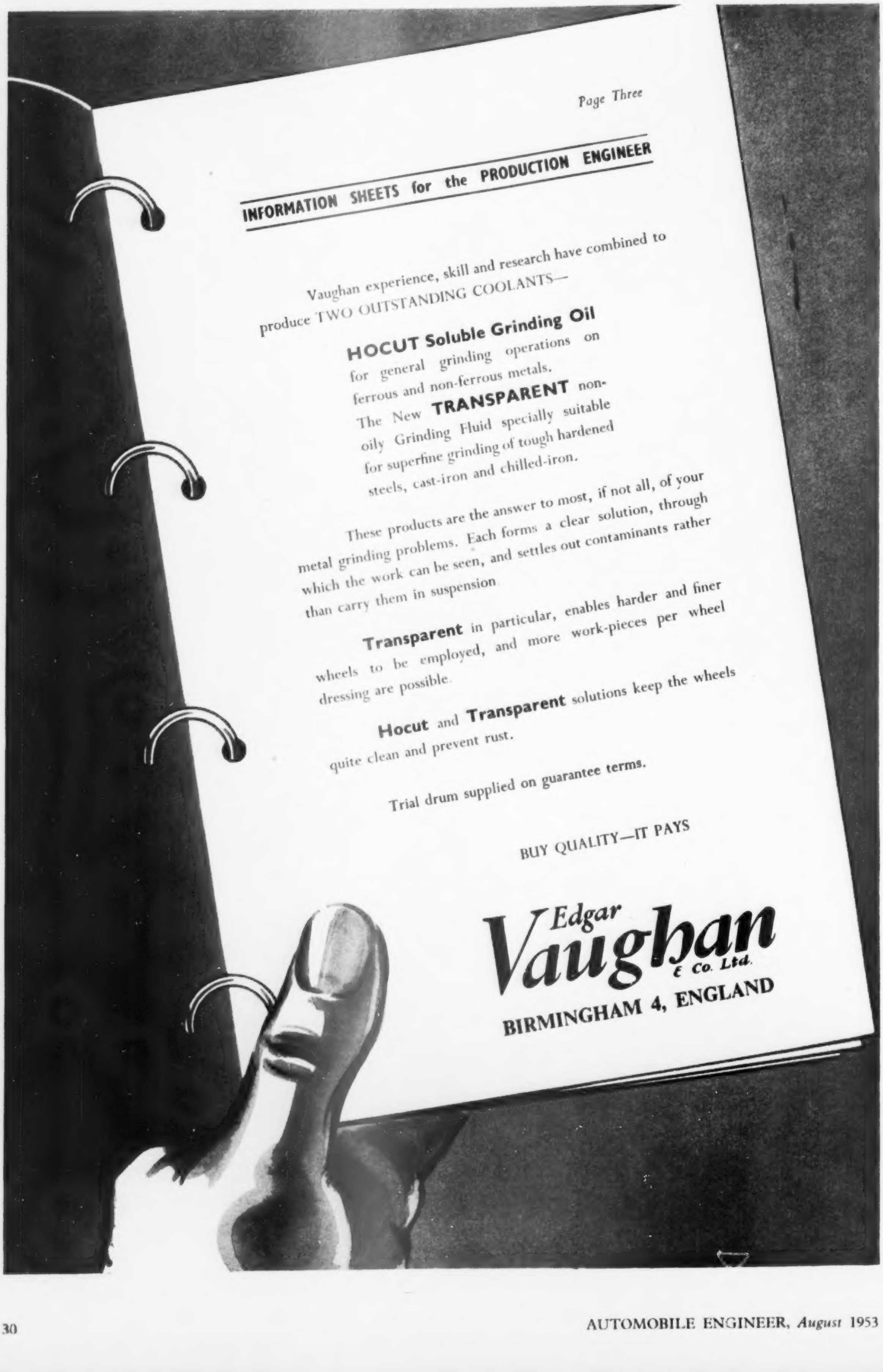


moulding fasteners
door casing fasteners
carpet fasteners
wire and cable fasteners


Page Three
INFORMATION SHEETS for the PRODUCTION ENGINEER
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for general grinding operations on ferrous and non-ferrous metals.
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Transparent in particular, enables harder and finer wheels to be employed, and more work-pieces per wheel dressing are possible.
Hocut and Transparent solutions keep the wheels quite clean and prevent rust.
Trial drum supplied on guarantee terms.
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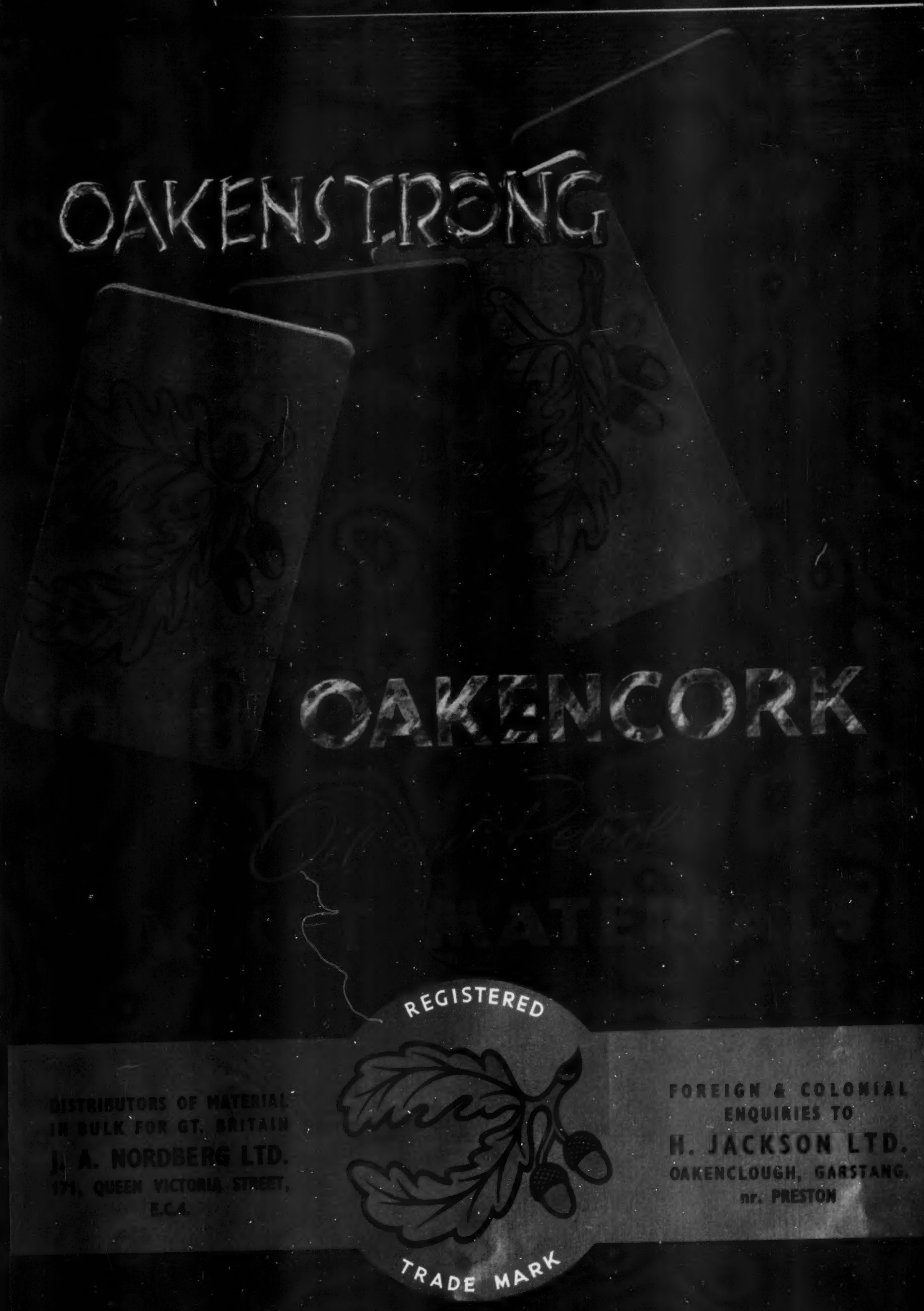
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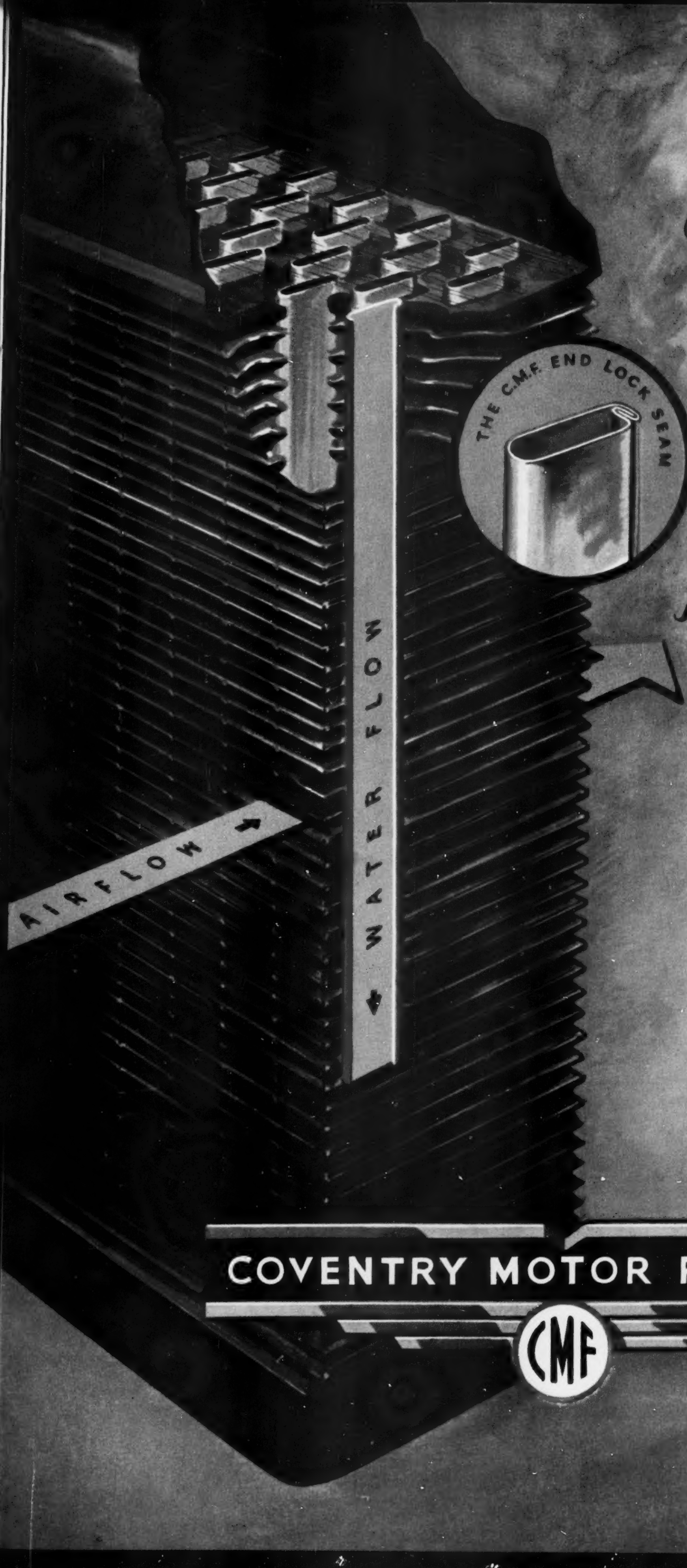
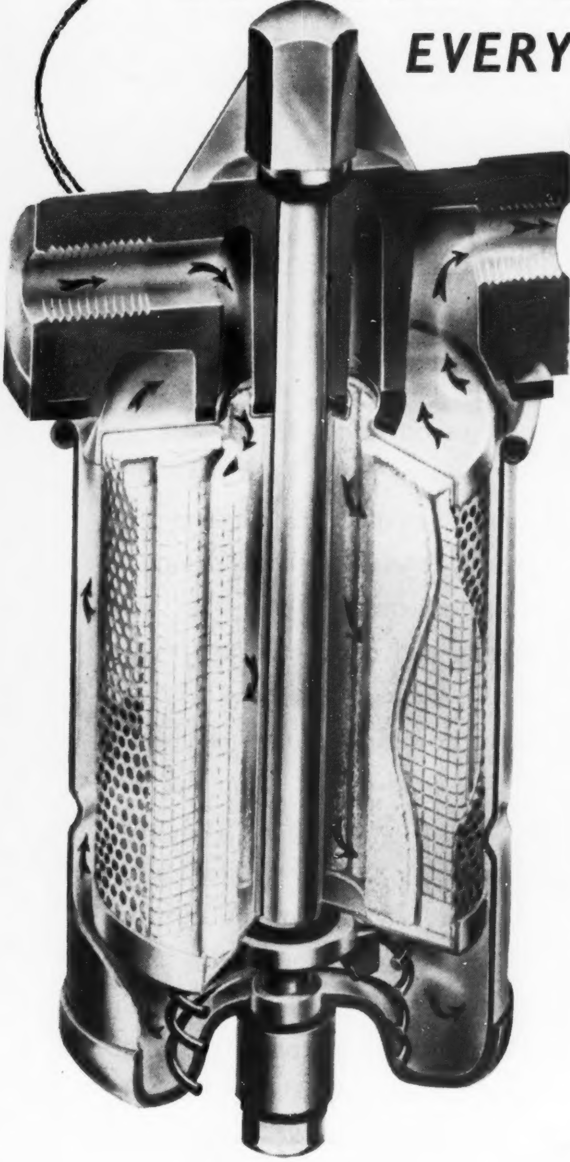

THE C.M.F. END LOCK SEAM
WATER FLOW
AIRFLOW

DO YOUR FILTERS ANSWER EVERY ESSENTIAL REQUIREMENT?



FILTERING ALL THE OIL ALL THE TIME

Only a full flow filter can give real engine protection. This fact is today almost universally acknowledged—yet even a so-called full flow filter must match up to all these requirements:—

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The VOKES element filters not only on the surface plane but, to a certain predetermined extent, *into the depth of the felt fabric itself.* This gives much greater efficiency in arresting *all* particles large enough to cause damage or wear and, even more important, ensures a build-up of sludge much slower than would be the case with an element providing surface filtration only.

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The policy of expendable elements may be satisfactory if spares are always available, but what happens if a machine in some remote place is prevented from working because no spares are available? VOKES filter elements can always be cleaned—a feature which ensures economy in the long run.

NON-RECIRCULATION OF TRAPPED SLUDGE

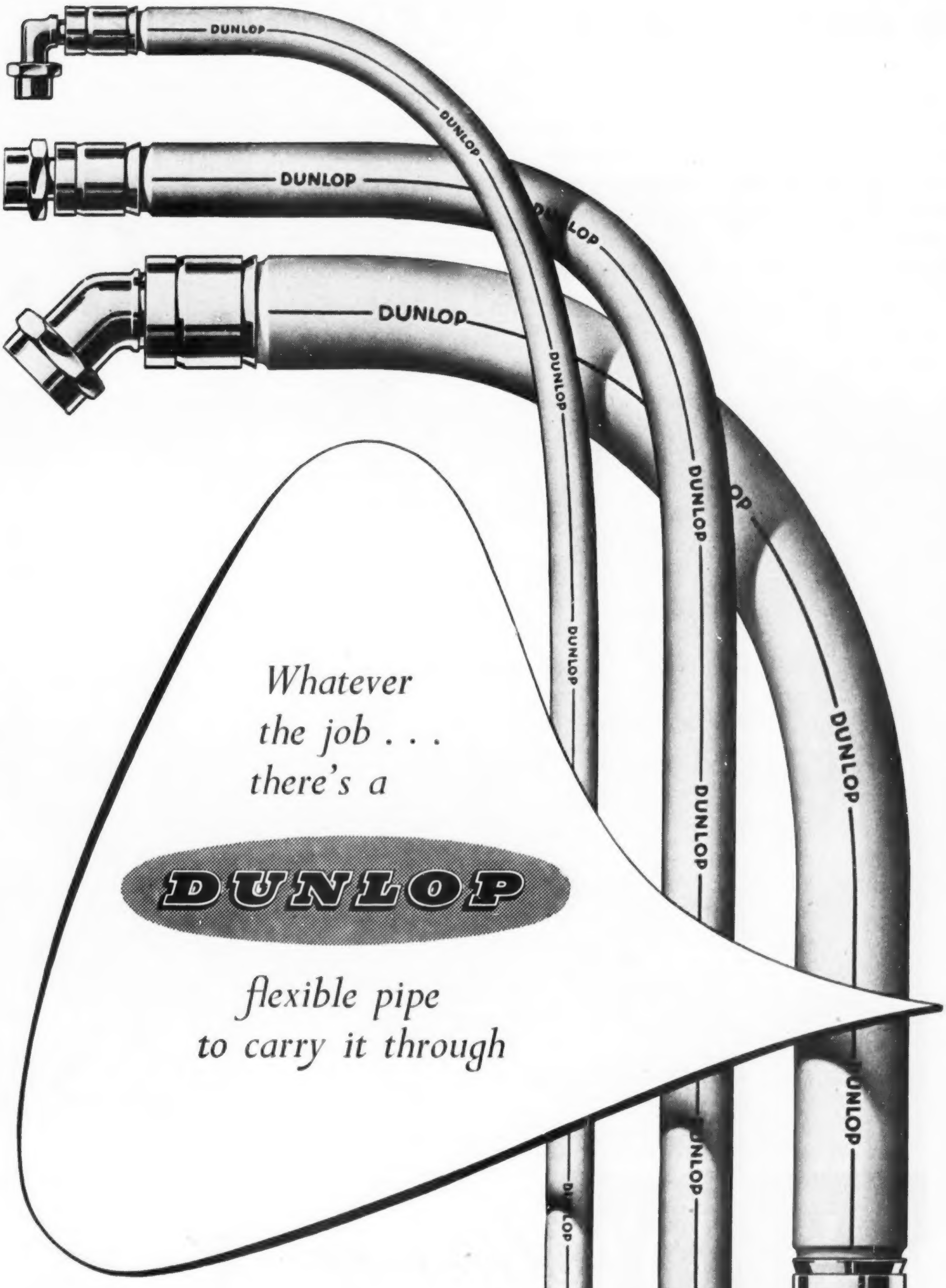
In the VOKES Principle, filtration is from the INSIDE to the OUTSIDE, which means that all the sludge is trapped in the centre of the filter element and cannot in any circumstances be recirculated.

Yes! . . . in thirty years, nothing has been found to improve on the basic principles of VOKES filtration for all-round efficiency.

VOKES *Pioneers of scientific filtration*

VOKES LIMITED · GUILDFORD · SURREY

VOKES (CANADA) LTD., Toronto. *Represented throughout the World.* VOKES AUSTRALIA PTY., LTD., Sydney


Whatever
the job . . .
there's a
DUNLOP
flexible pipe
to carry it through

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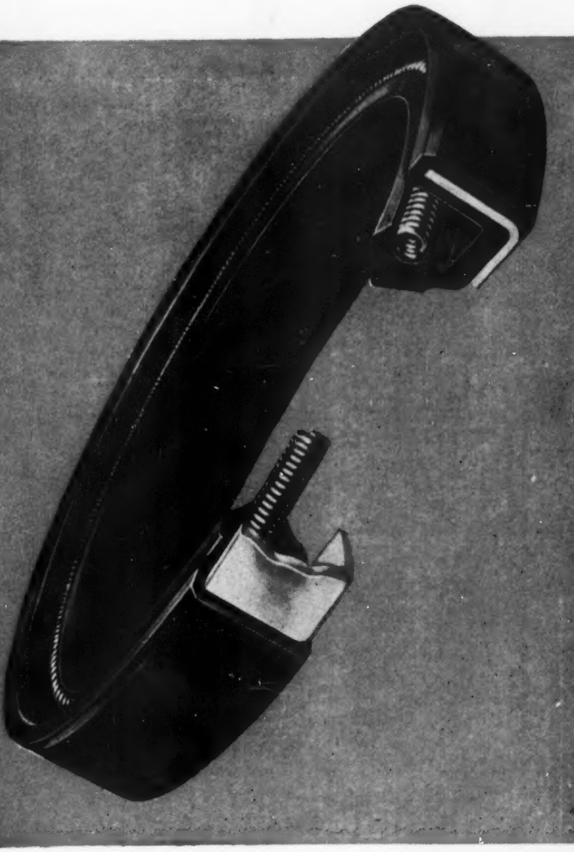
THE GACO M.I. OIL SEAL

British Patent No. 479743

The Scientifically Designed Oil Seal for Rotary Shafts. A Gaco Product.



1. Deep rim affords perfect Spring Retention.
2. Large Face Angle brings Sealing Edge well below spring.
3. Knife edge Contact at Sealing Point.
4. Sealing Point stiffened against local deformation by large included angle.
5. This section stiffened to prevent deformation under load.
6. Shaft angle gives adequate clearance.
7. Flexible Web.
8. Gaco Skin affords better fluid tight fit in Housing.

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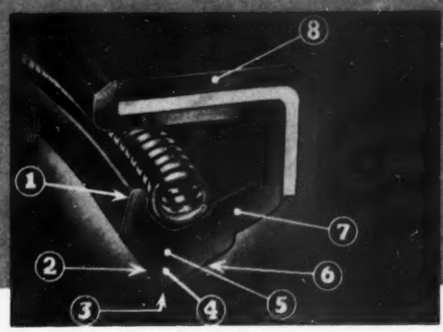

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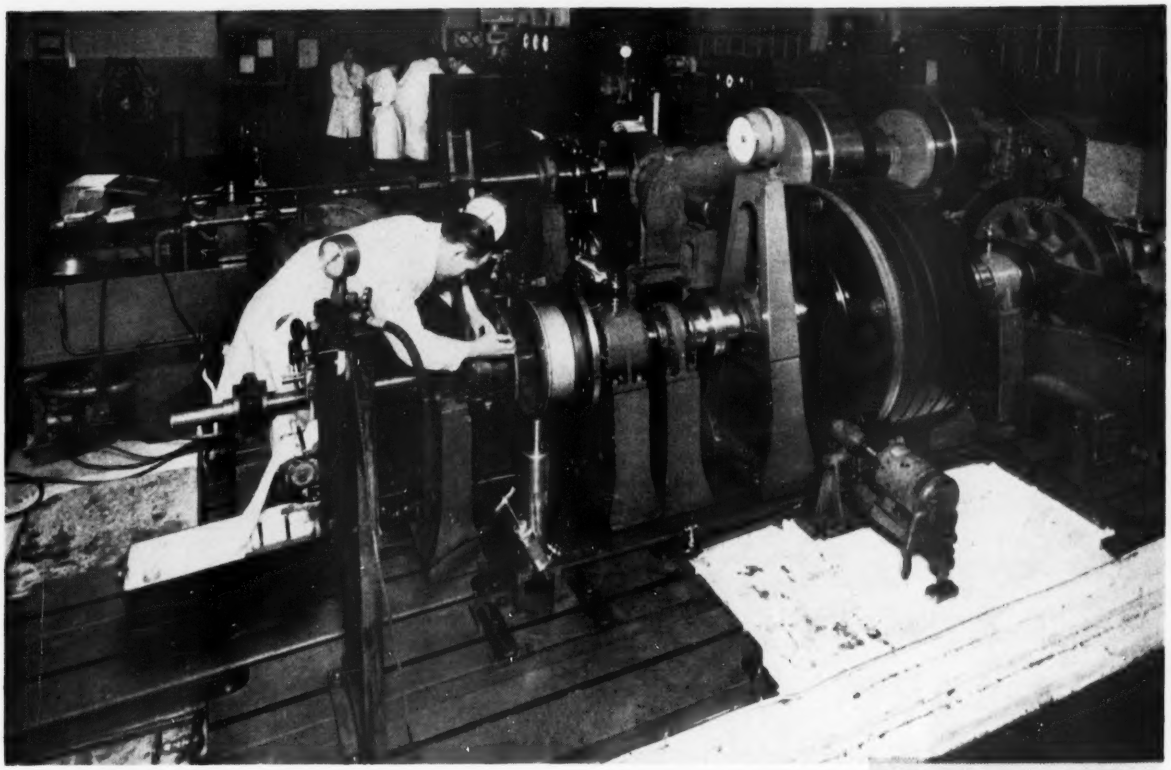

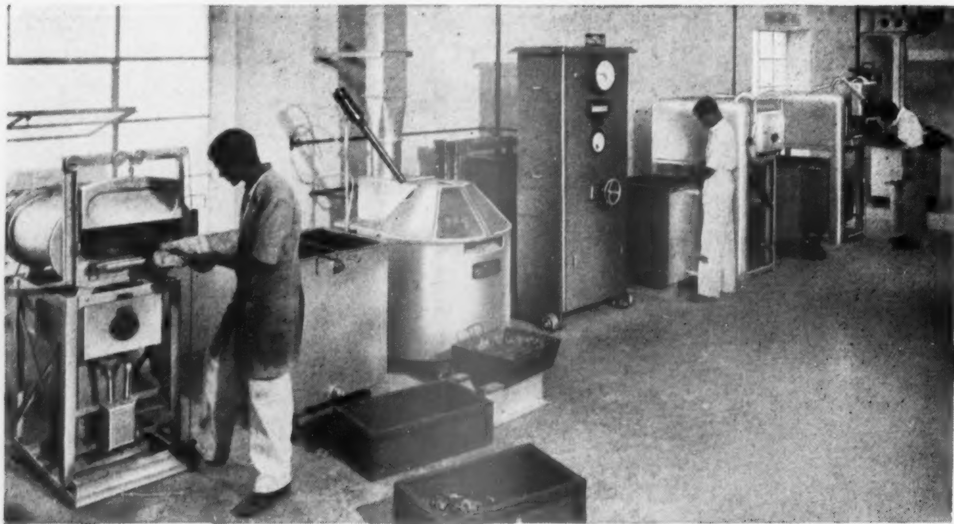

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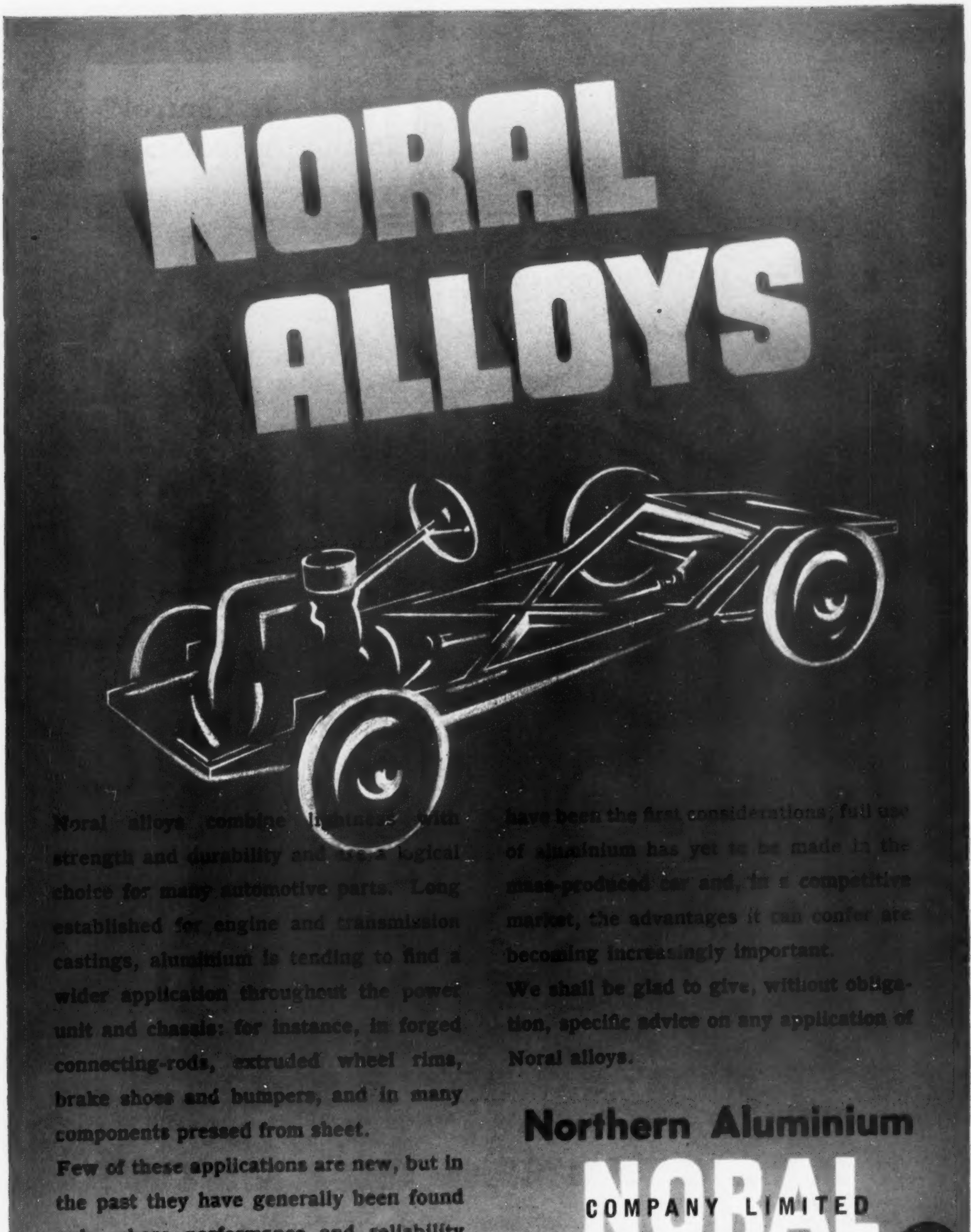
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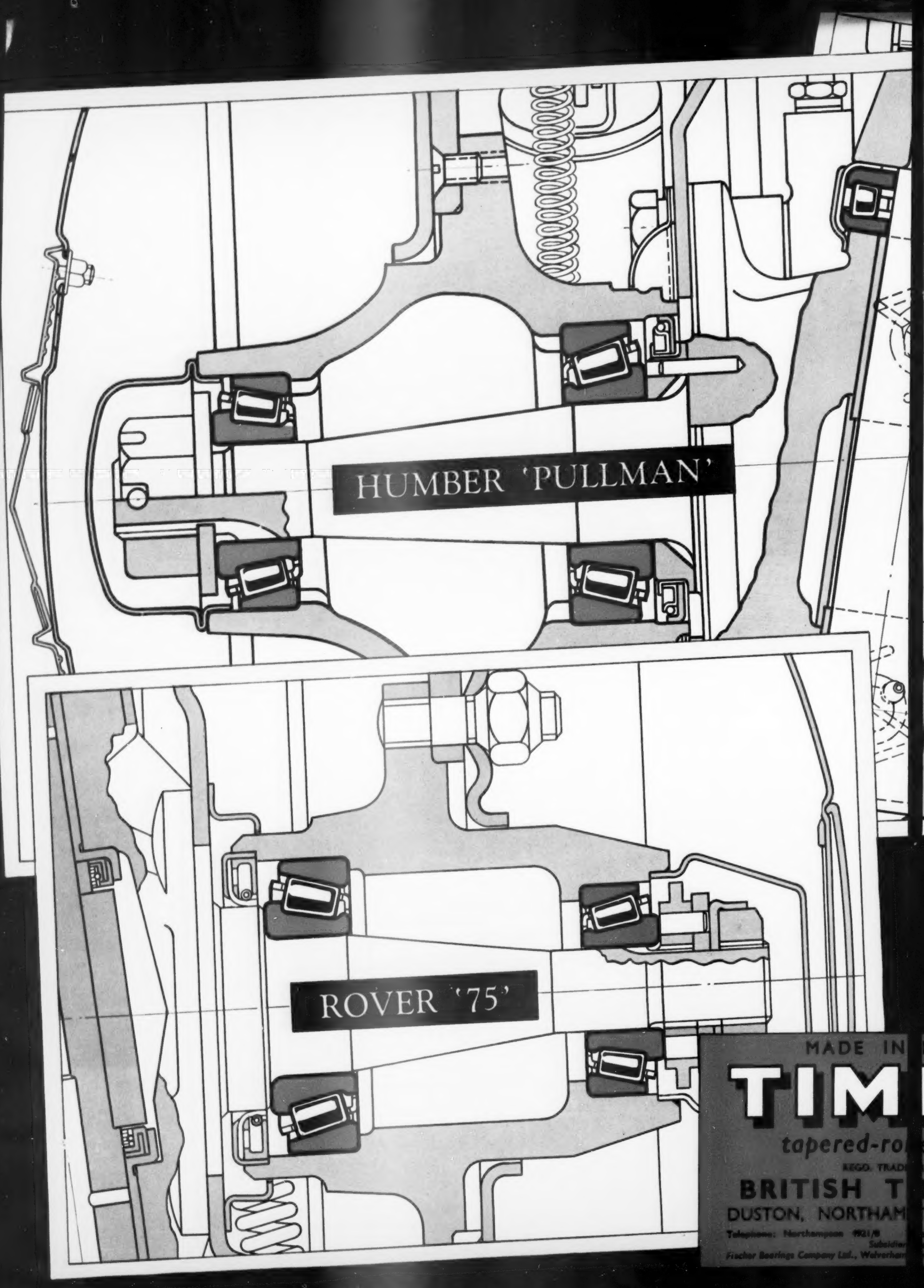

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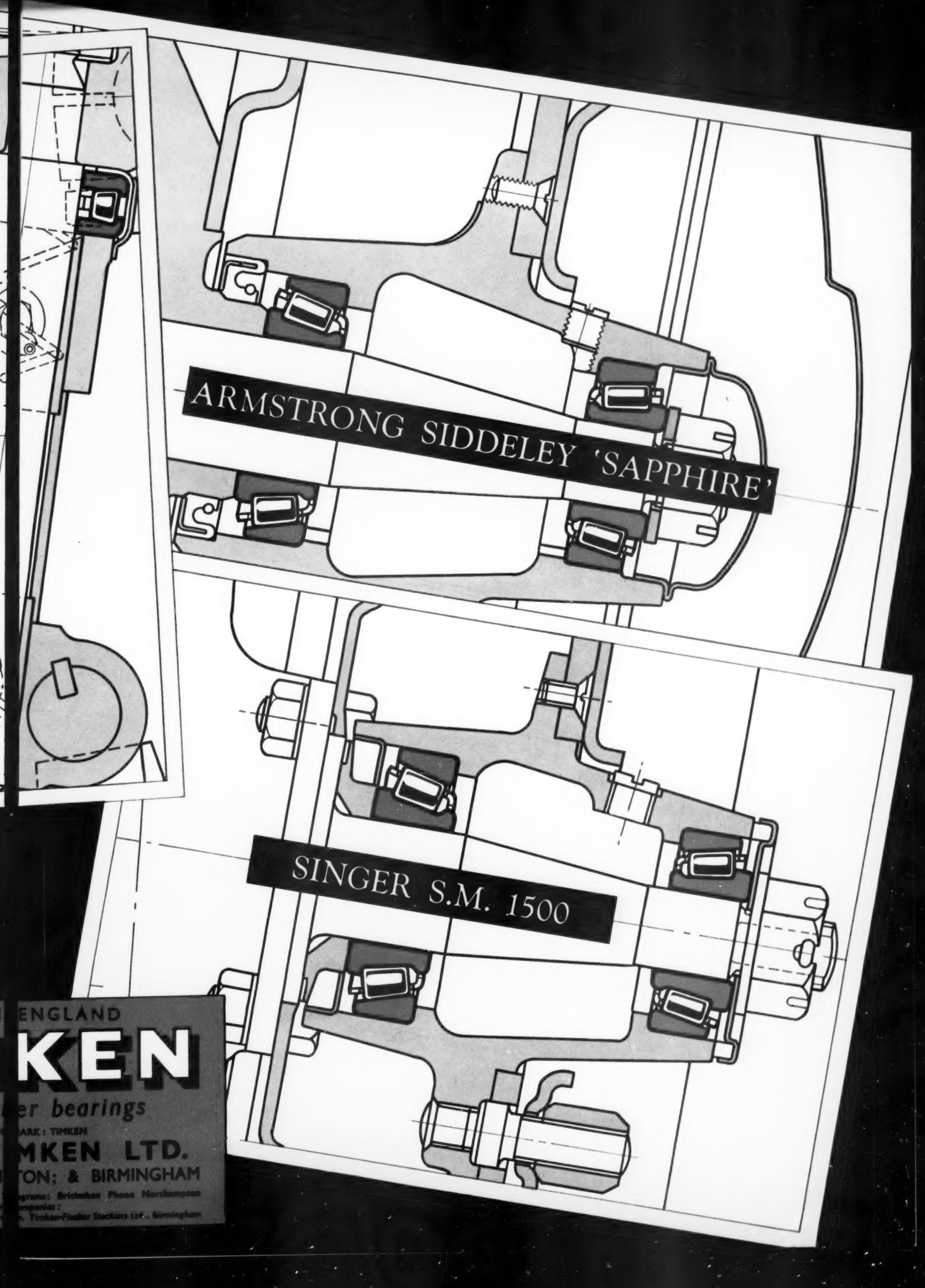
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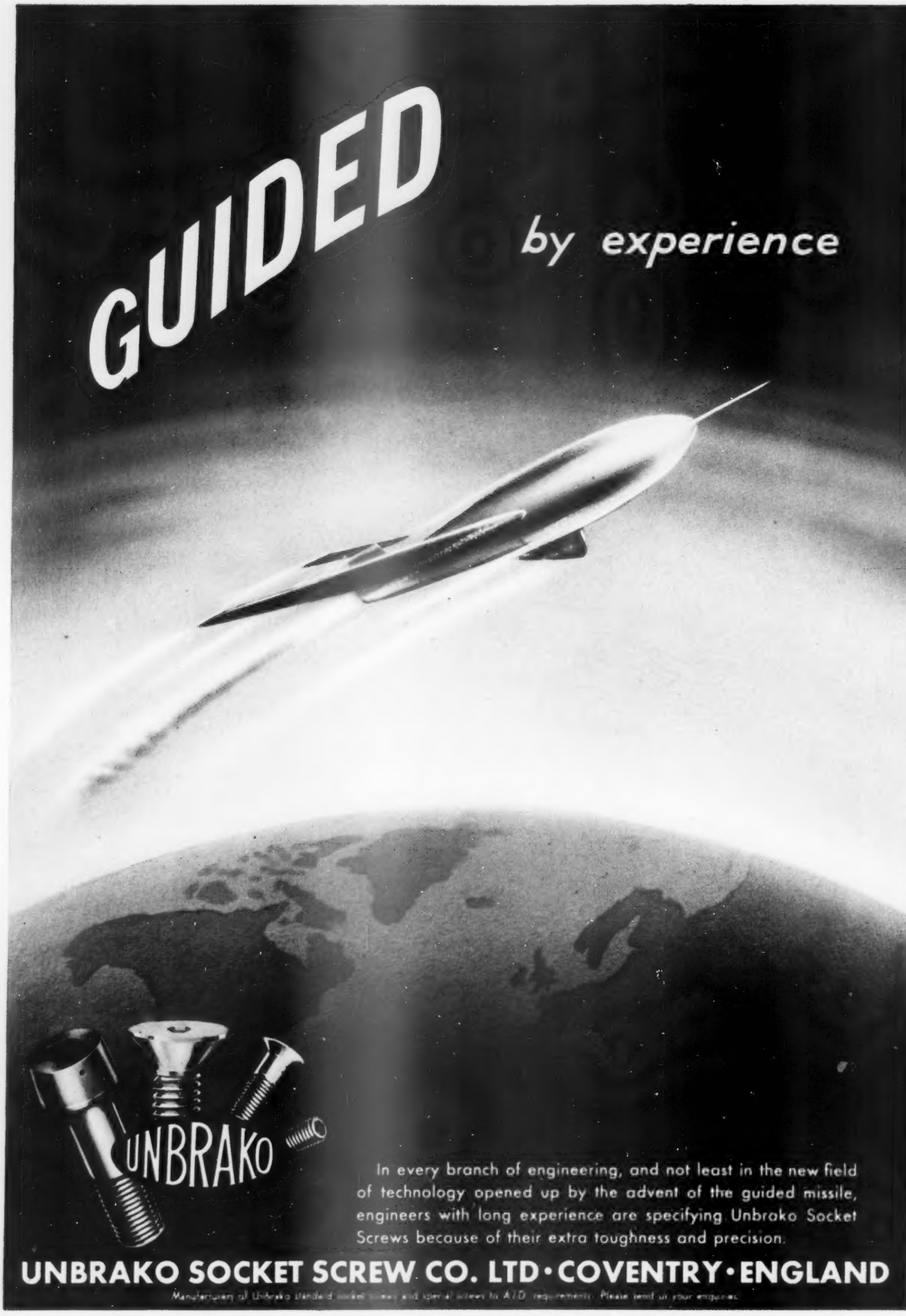
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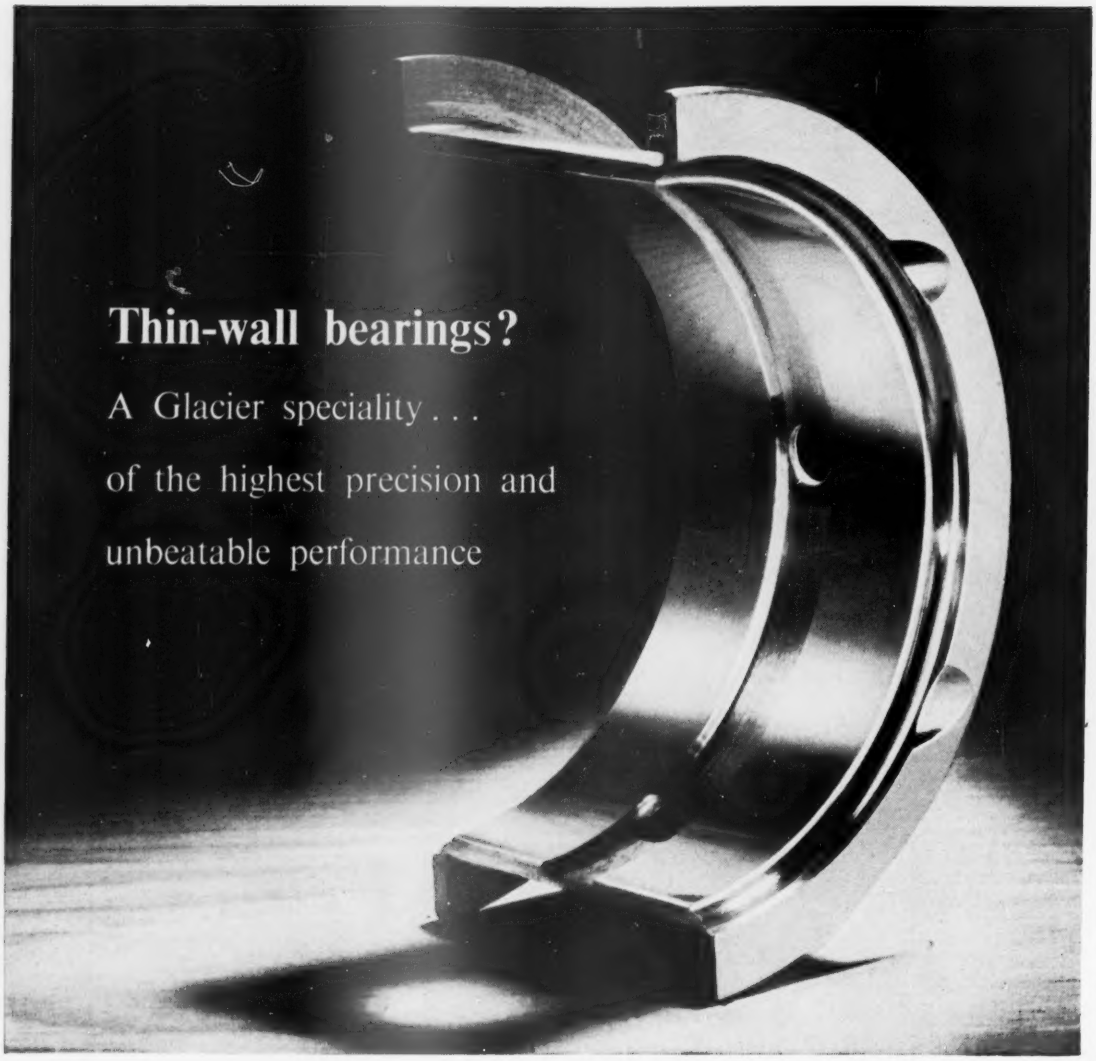

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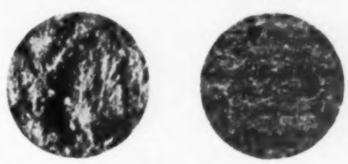

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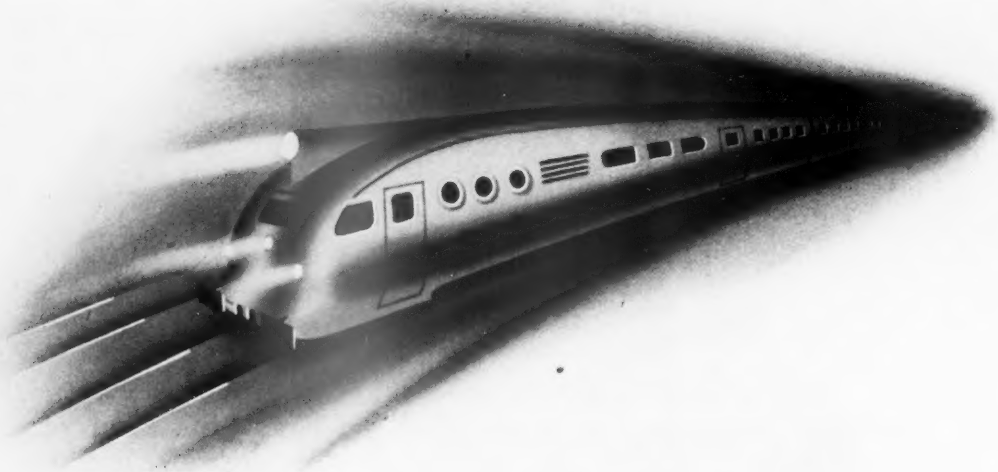


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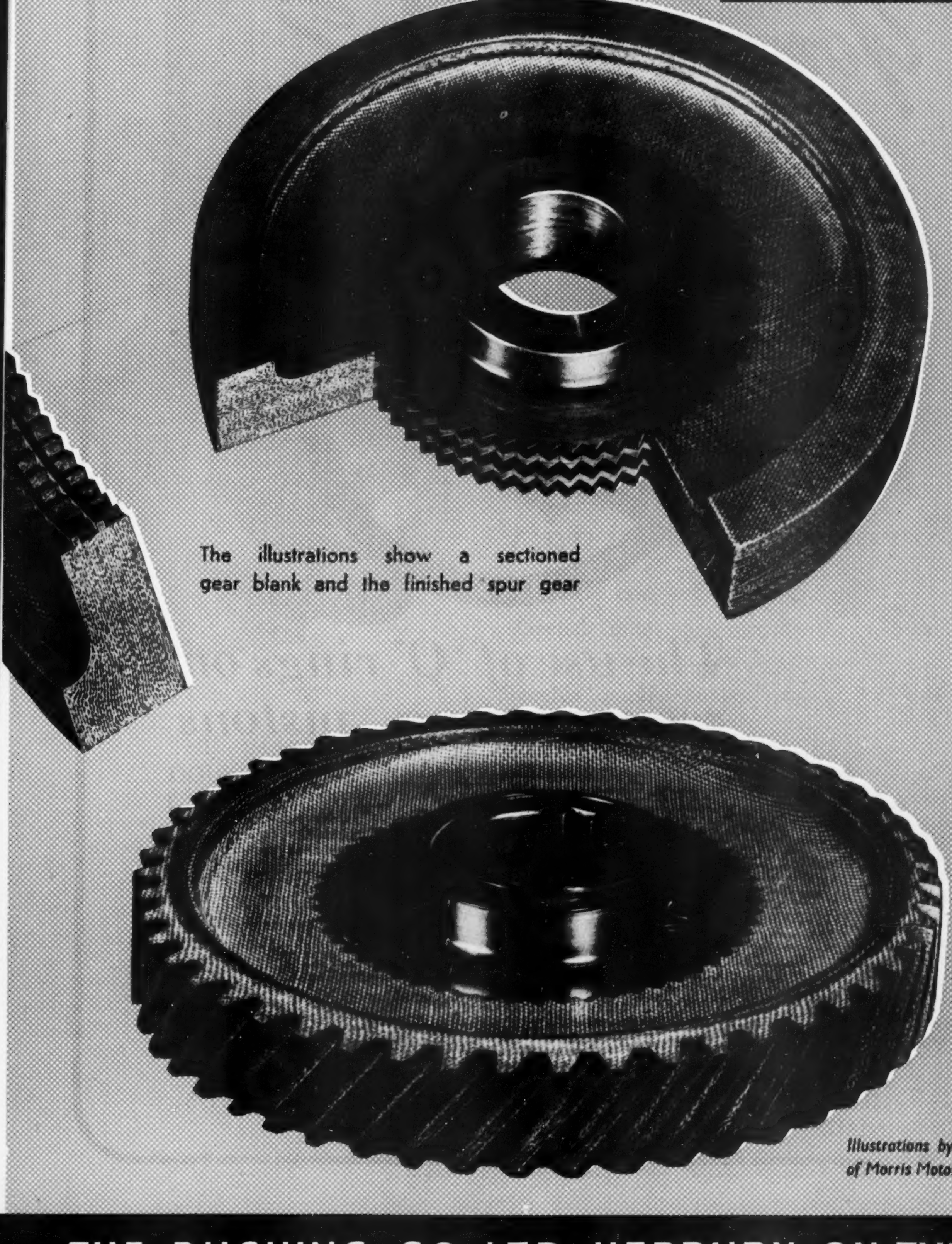

The illustrations show a sectioned gear blank and the finished spur gear

Illustrations by courtesy of Morris Motors Limited

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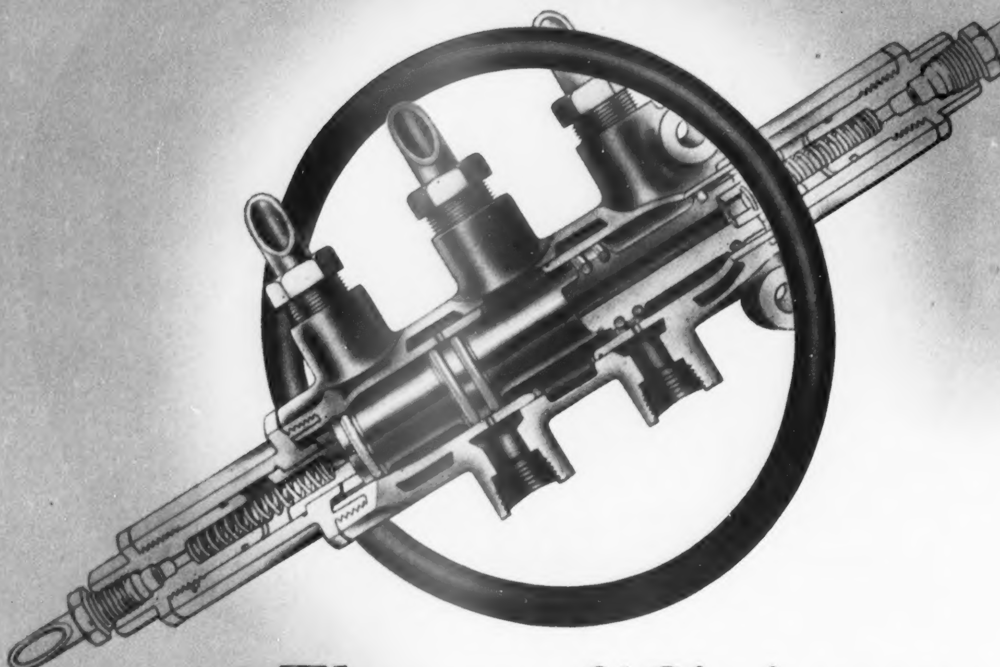

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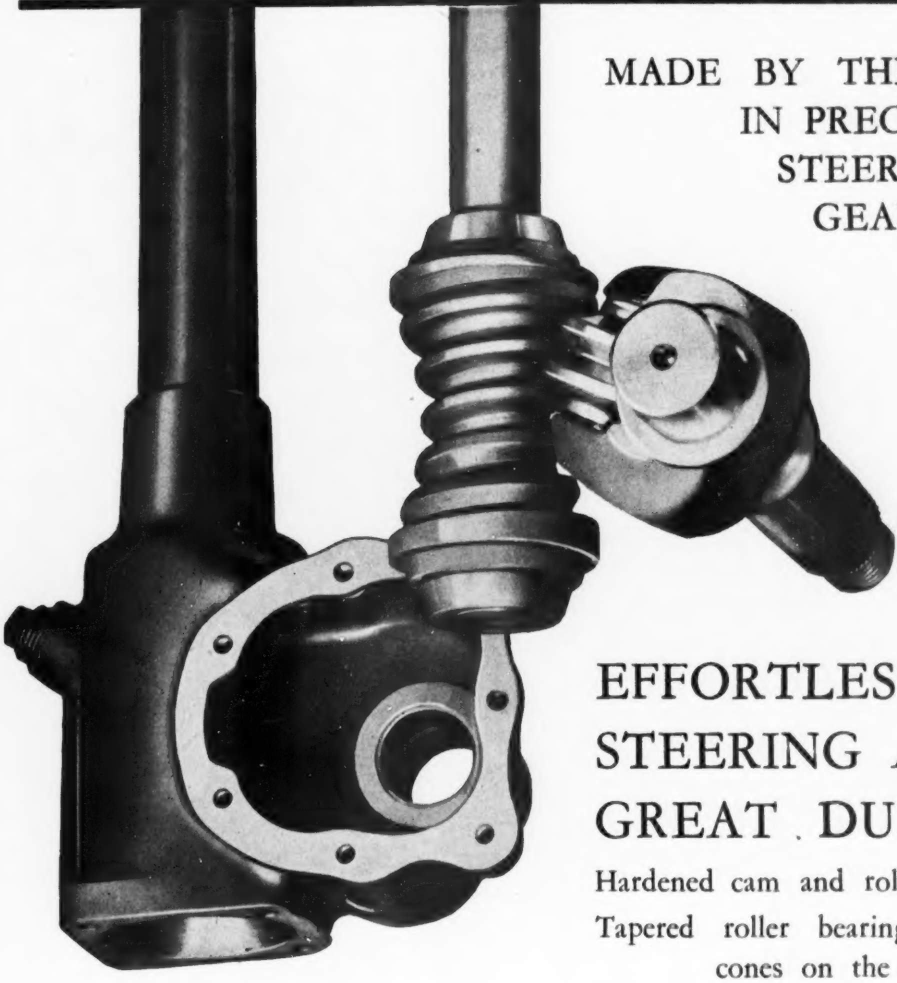

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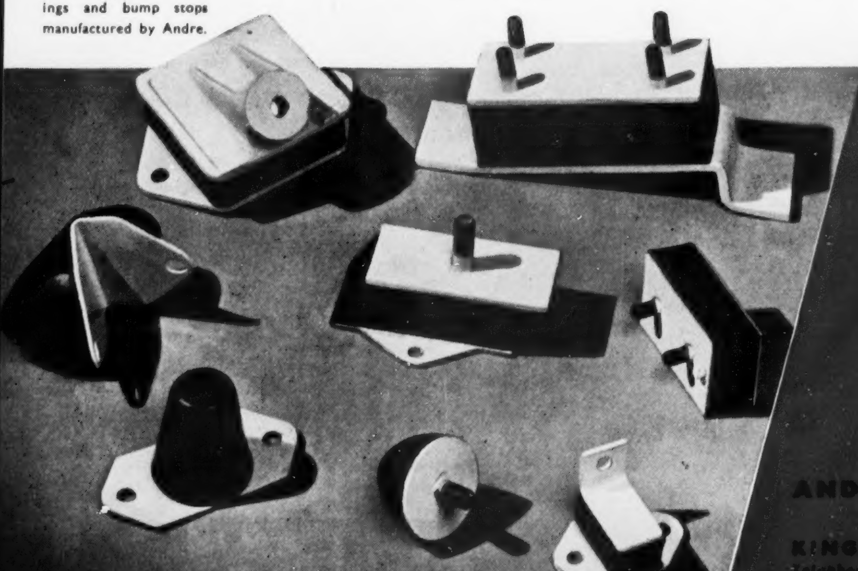

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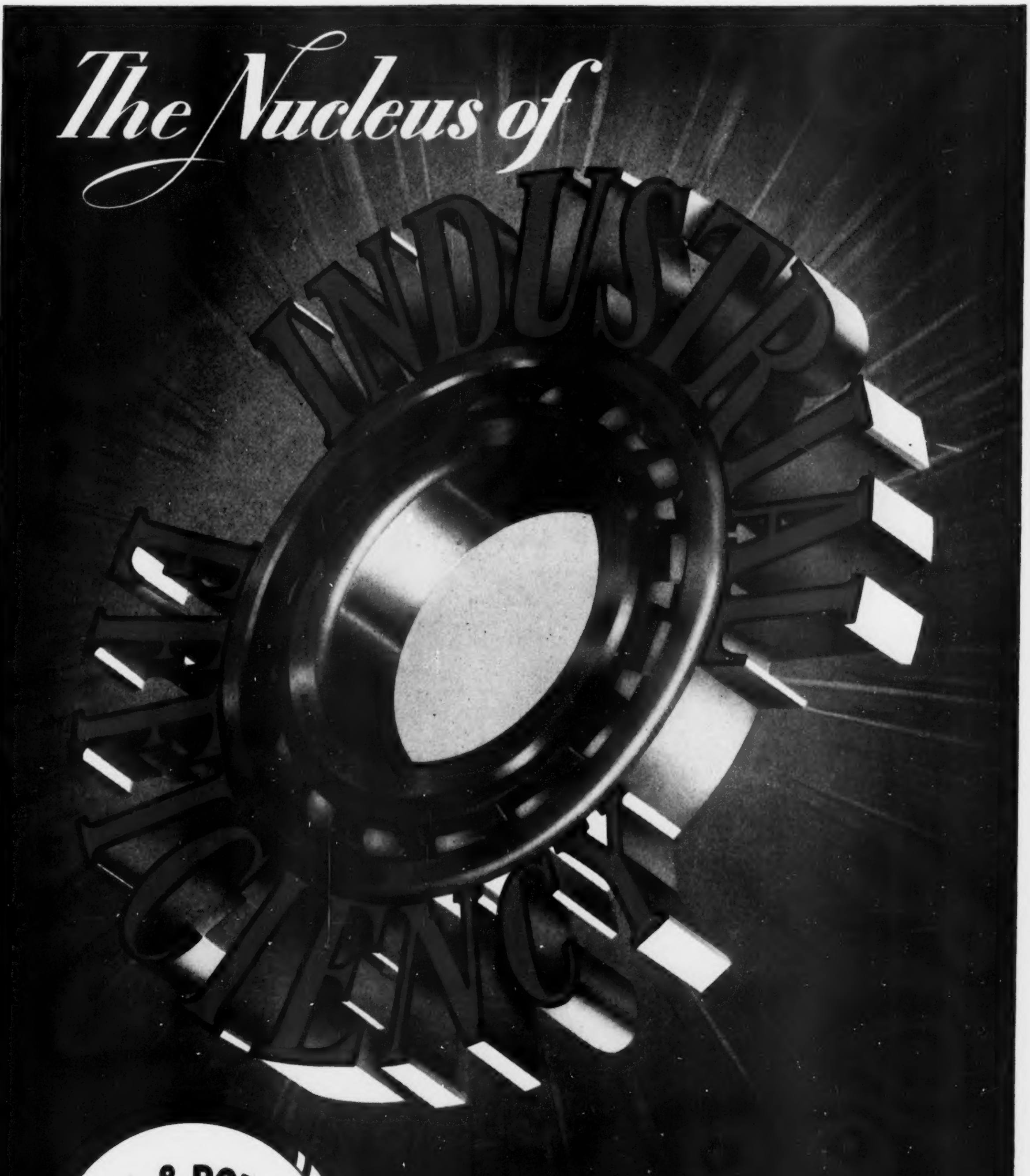

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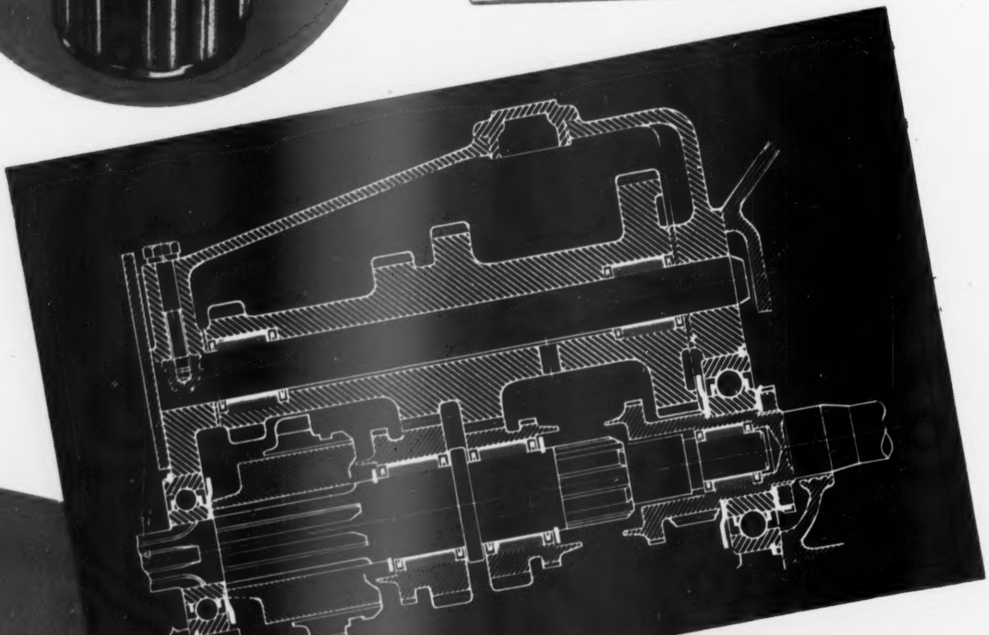

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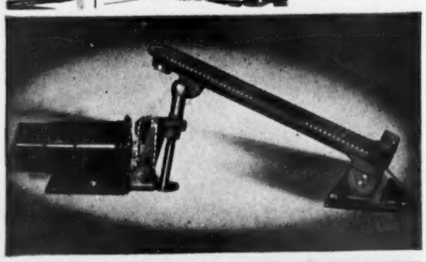

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Spot Disc Brakes

THERE can be no doubt that the decisive Jaguar victory at Le Mans must rank as an astonishing performance. It is difficult to assess the value of such a success to British prestige, but it will almost certainly lead to a further increase in the sales of this car in the all-important dollar markets. The performance of this 3½ litre power unit was the outstanding feature of the race. Once again the engine demonstrated superb qualities of reliability and durability under most punishing conditions. We recall with satisfaction our published comments when the model was introduced.

A highly significant factor in the race was the manner in which the spot disc brakes, fitted to the Jaguar cars, enabled the drivers to exploit the power of the vehicle to the full. This opportunity of utilizing in the maximum possible degree the power potential of the engine, permitted the maintenance of a speed standard that shook other contestants. Even so, given drivers equal to the occasion, power and overall durability still remain the dominating factors. After all, the winning cars were driven harder than many of the rivals that cracked up, but they stood up magnificently to the most gruelling treatment.

Manufacturers of heavy and powerful vehicles will now be more keenly interested than ever in the spot disc brakes. They now seem to have reached a stage when, before long, they should make an appearance on certain types of production vehicles. We cannot claim an intimate knowledge of the work that has been carried out in developing these brakes, but there would appear to be two outstanding problems; firstly, that associated with the small size of the shoe, and secondly the old problem of heat dissipation.

In the case of the first point, it appears to be inevitable that the much reduced lining surface must lead to more rapid wear than is associated with the conventional drum lining. This could, of course, be partly overcome by fitting considerably thicker lining material. In the final resort, it could be overcome by arranging for extreme simplicity of replacement. In short, the brakes could be so designed that the pads could be slipped in and out in a matter of moments. They would be regarded as expendable material.

For a given degree of generated heat the dissipation characteristics of the spot disc brake will be better than those of the drum type, since it is not so shrouded against the cooling effect of the ambient air. On the other hand, for a given braking effort, the spot disc brake will probably have a higher localized temperature. Fortunately, the disc brake is not so susceptible to distortion as the drum and high temperatures are not so serious in this respect. Whether the cooling effect of the air stream will be sufficient to prevent the transmission of excessive heat still remains to be seen.

It appears certain that pressure between the shoe and the disc will be more uniform over the contact area. As a result, the braking should be more stable, and therefore more constant. Consequently, wear should be uniformly distributed over the whole contact area. There are other subsidiary advantages that make the spot disc scheme very attractive, and there can be little doubt that further development work will overcome any problems that may still remain to be solved.

The basic principle is, in any event, so sound that rapid development is obviously worth while. To appeal to a wider market, the spot disc brake will, of course, have to be competitive in cost, light in weight and adaptable to the various commercial applications that will be demanded. For use on heavy vehicles, fast coaches and large, high speed private cars the demand is obvious.

Titanium

ALTHOUGH more than seventy metallic elements are known today, very few are used by engineers. In the British Standards specification no more than twenty-six metals are mentioned, and most of these are used only in small quantities to form alloys of, what may be regarded as the four basic engineering metals, iron, aluminium, copper and zinc. Some of the metallic elements mentioned in the standards specification are, in fact, included only because their presence in certain alloys is detrimental.

Many of the metallic elements are not used in engineering because they are available only in small quantities in the earth's crust, but this is not true of titanium. There is, in fact, about sixty times more of this element than there is of copper. The sole reason why titanium has not been more widely used hitherto is that its production from ore is still very costly. Experience with aluminium suggests that cost will not always remain an insuperable bar to wider use.

One hundred years ago aluminium cost about £125,000 a ton, and sixty years ago world production was only 750 tons per annum. Although it was not until 1940 that

W. J. Kroll showed how to produce ductile titanium in quantity, world production to-day is certainly more than 750 tons per annum and the cost is about £8,000 per ton. This price is not, of course, in any sense a guide to what the cost may be in five, or even three years' time. In fact, the rapid progress that has been made in the short period since production on anything other than a laboratory scale has been possible suggests that the metal may well be in general use in the not so distant future.

Some conception of the applications for which it is suitable may be obtained from its physical properties. The melting point is high, 1,725 deg C, and in the molten state the metal is chemically intensely active. In contrast, its corrosion resistance at normal temperatures is such that it may be used to replace stainless steel, and for equal dimensions, a titanium part will weigh about 50 per cent less than a stainless steel part.

Commercially-pure titanium has a 0·2 per cent proof stress of about 33 ton/in^2; the ultimate strength is approximately 39 ton/in^2 with an elongation of 22 per cent. The density of the material is about 57 per cent that of steel and 1·7 times that of aluminium. In view of the fact that pure iron has an ultimate strength of appreciably less than 20 ton/in^2, while some alloys containing more than 95 per cent of iron have an ultimate tensile strength of more than 100 ton/in^2, it seems likely that alloys of titanium of exceptionally high tensile strength may be developed.

A considerable amount of research has been devoted to alloying titanium with other metallic and non-metallic materials. Although high tensile strengths may be obtained with single alloying elements the elongation is generally drastically reduced. Better properties, so far as elongation is concerned, are obtained when two or more alloying elements are added. For example, when 5 per cent Cr, 3 per cent Al and 0·5 per cent C are added, the ultimate tensile strength is 70 ton/in^2, and elongation 10 per cent.

Such alloys retain much of their strength at temperatures at least as high as 400 deg C. This characteristic, together with their light weight and corrosion resistance, makes them ideally suited for pistons and other reciprocating parts subject to high temperatures. However, apart from the present day cost of the material, there is one serious objection to its use for pistons. This is that the bearing properties of the material are very poor, probably due to its intense chemical activity at the very high temperatures obtained locally when two pieces of metal are rubbed together.

Nevertheless, it is possible that the tendency to pick-up may be overcome by suitable treatment of the surface. For instance, titanium and its alloys may be given a hard surface by anodizing. Alternatively they may be case hardened by heating in oxygen or nitrogen, for the metal forms alloys with these gases, which go into solid solution.

It is not suggested that orders should immediately be placed by component manufacturers for a ton of titanium for experimental purposes; the material has not yet been developed to a point where it is even approaching the stage of being a sound commercial proposition. Nevertheless, developments should be watched carefully.

20 M.P.H.

NORMALLY, we are not concerned in these columns with legislation governing the use of vehicles on the road; that is much more the province of journals such as *Autocar*, *Bus and Coach* and *Motor Transport*. There is however, reason for cavilling at the recent decision to maintain the statutory limitation of 20 m.p.h. for heavy road vehicles. In our opinion this decision is bad both economically and psychologically.

Heavy road vehicles are of major importance in the transport of goods, and handling charges are in many cases a substantial proportion of total costs. We do not suggest that an increase in the permissible speed of heavy vehicles would lead to a substantial reduction in costs, but the position to-day is such that, any marked saving in costs can come only through the cumulative effect of many minor savings.

Psychologically, the limitation to 20 m.p.h. is bad because it is so frequently disregarded, and disregard for one law only too frequently brings other and better laws into contempt. As all road users know, the vast majority of heavy vehicle drivers, if not all, habitually exceed 20 m.p.h. Nor is it possible that the limitation can be universally enforced. Would it not be wiser to recognize this?

In some quarters there is a belief that the limitation serves the interests of road safety. We respect the sincerity of those who hold such thoughts, but logic is against them. Road conditions vary so greatly that there must be a very wide variation in safe speeds. In fact, it is arguable that a higher speed limit for heavy goods vehicles would improve the safety factor on long distance work since the driver would have greater latitude in maintaining schedule.

CONTENTS

COMPACT ENGINES

Some Comments on Design for Installations Where Space Limitations are of Prime Importance

IT is probably true that during the last half century every conceivable layout for internal combustion engines has been tried, and to-day major dimensional economies can be effected only by the adoption of somewhat unconventional layouts. Nevertheless, appreciable advantages may also be derived from a combination of a number of detail economies, such as reducing the height of valve gear, unconventional sump arrangements, etc.

It has now been firmly established that the four cylinder in line arrangement with a wet sump is the least expensive engine layout suitable for private cars. Admittedly, cheaper designs may be adopted but they all have serious disadvantages. In general, any attempt to depart from the conventional layouts to produce an acceptable compact engine automatically increases the cost.

However, there is a tendency to consider the cost of the power unit only, and such an attitude can be misleading. A large engine, apart from encroaching on the space otherwise available for passengers, requires a correspondingly large bonnet and longer chassis frame for its accommodation, and if it is heavy, a more substantial structure may be needed to support it. Thus, when cost is being considered, the installation as a whole must be examined.

In designing an engine for a particular installation there are two important conflicting requirements to be considered. One is that the general shape of the space into which it is to be fitted may call for a flat engine, or perhaps a narrow vertical arrangement. The other and often overriding requirement is that it should be suitable for installation in all models of a similar size that the company is likely to produce for the subsequent fifteen years or so, otherwise tooling costs may be high. This does not mean that the design should be frozen for that period of time; but normal development leading to improvement must be carried out within limitations imposed by the basic layout of the engine.

In private cars, compactness ranks among the most important requirements only in a very small vehicle. In larger ones styling considerations call for the incorporation of a bonnet of such a length and size as will give balance to the overall artistic conception. This is usually adequate to house a conventional in line engine. Nevertheless, the smaller the engine the greater the latitude in the general design of the vehicle. In commercial vehicles compactness is again of prime importance, and layout is often largely determined by the shape of the space into which the unit is to be fitted. An outstanding example of the marked effect of this influence is the widespread adoption of the flat, underfloor engine.

Small cars

Unfortunately, among the design requirements of small cars, powered by engines of about 900 c.c. or less, low cost ranks before compactness. There are few people, if any, who will pay more for a car simply because it is small; in fact, they expect to pay less. However, as has already been mentioned, the criterion is the cost of the whole installation and not just of the engine. A number of the continental manufacturers have adopted somewhat unconventional layouts, and they have been successful. Therefore it would appear that a study of the factors involved might be rewarding.

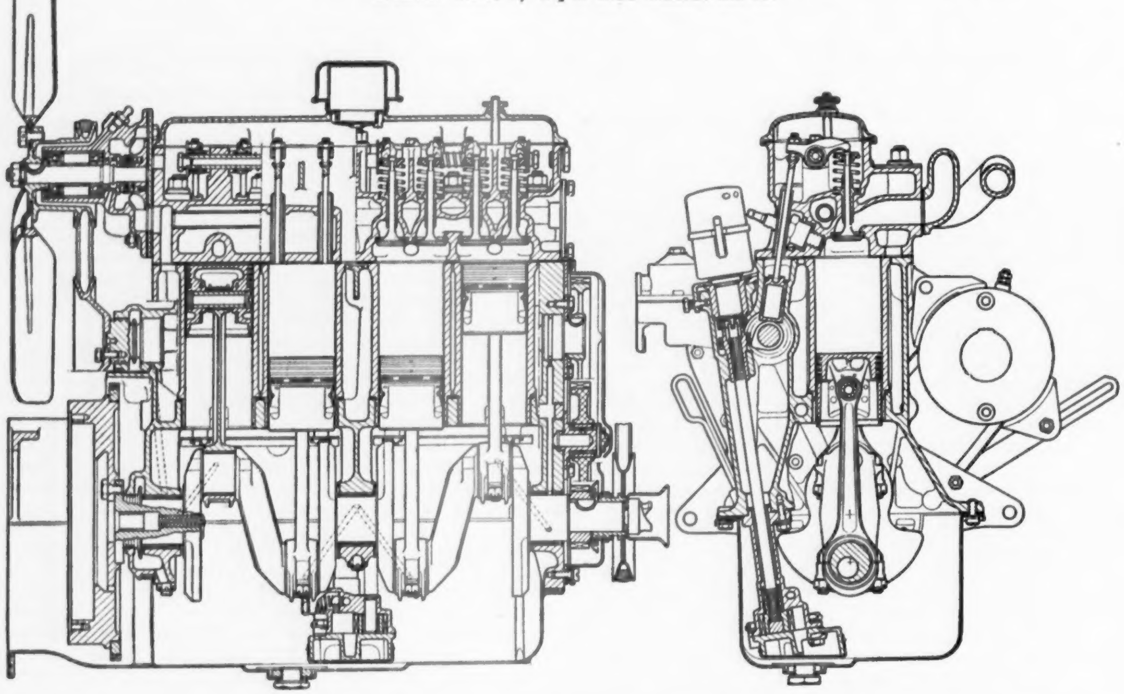
There is much to be said for grouping the engine and transmission either at the front or at the rear end of the car. In this way, not only are the expense and weight of the long transmission avoided, but also the body floor may be both flat and low. The elimination of the propeller shaft tunnel gives more room inside the car and at the same time makes the structural design problem easier.

Special considerations govern the design of power units for either front wheel drive or rear engined small cars. So far as front wheel drive is concerned, the bonnet inevitably is unduly long unless a two cylinder engine, preferably in the horizontally opposed twin form, or a horizontally opposed four cylinder unit is employed. There are two reasons for the preference for a horizontally opposed layout. First, the overall length is less, and secondly the balance is better, as may be seen from the following list of

On the Citroen 2CV, the dynamo is driven directly by the front end of the crankshaft, as may be seen from this illustration of the sectioned engine

One of the disadvantages of air-cooled engines for motor cars is that a relatively large capacity blower must be incorporated for cooling purposes

THE 750 c.c. RENAULT ENGINE
The overall dimensions of this unit are 21¾ in long, by 20½ in deep over the sump drain plug and breather filter, by 12½ in wide without the fan



unbalance forces and couples for engines with 180 deg crank throws.

Vertical twin: primary couple and secondary force.

Opposed twin: primary and secondary couples.

Vertical four: secondary force.

Opposed four: secondary couple.

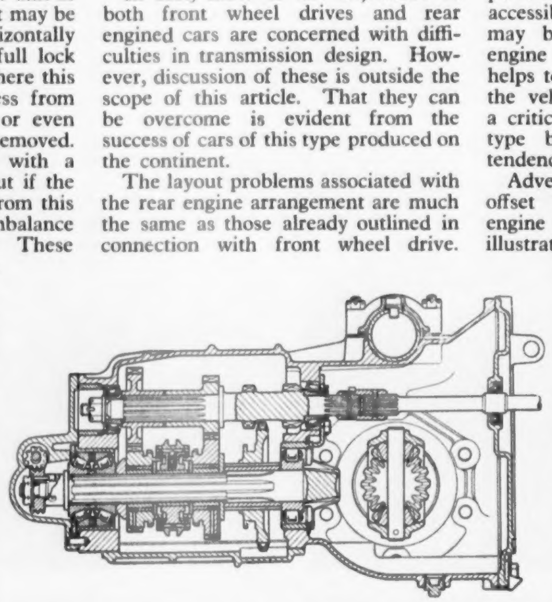
The couples are not so great in the opposed engines because of the decreased length of this type of unit as compared with in line ones. It may be difficult to accommodate a horizontally opposed engine between the full lock positions of the wheels but, where this arrangement is possible, access from the side may be satisfactory or even good when a front wheel is removed. Firing impulses are uneven with a 180 deg crank arrangement, but if the 90 deg crank, which is better from this point of view, is adopted, unbalance primary forces are introduced. These forces may be partly balanced by suitable adjustment of the rotating masses on the crankshaft. However, lateral unbalance forces are then unavoidable, even though they are only about half the magnitude of the primary forces before these were partly balanced.

Another requirement of the small front wheel drive car is that the front wheels must be as far forward as possible. This is to avoid encroachment by the wheel-arches on the toeboard width. As a result, it is almost essential with a flat four, and certainly desirable with a flat twin cylinder arrangement, to interpose the final drive between the clutch and the gearbox. With this arrangement a long slender shaft is necessary to transmit the drive from the clutch to the gearbox, and torsional vibration and fatigue troubles may be experienced unless the appropriate measures are taken at the design stage for their avoidance.

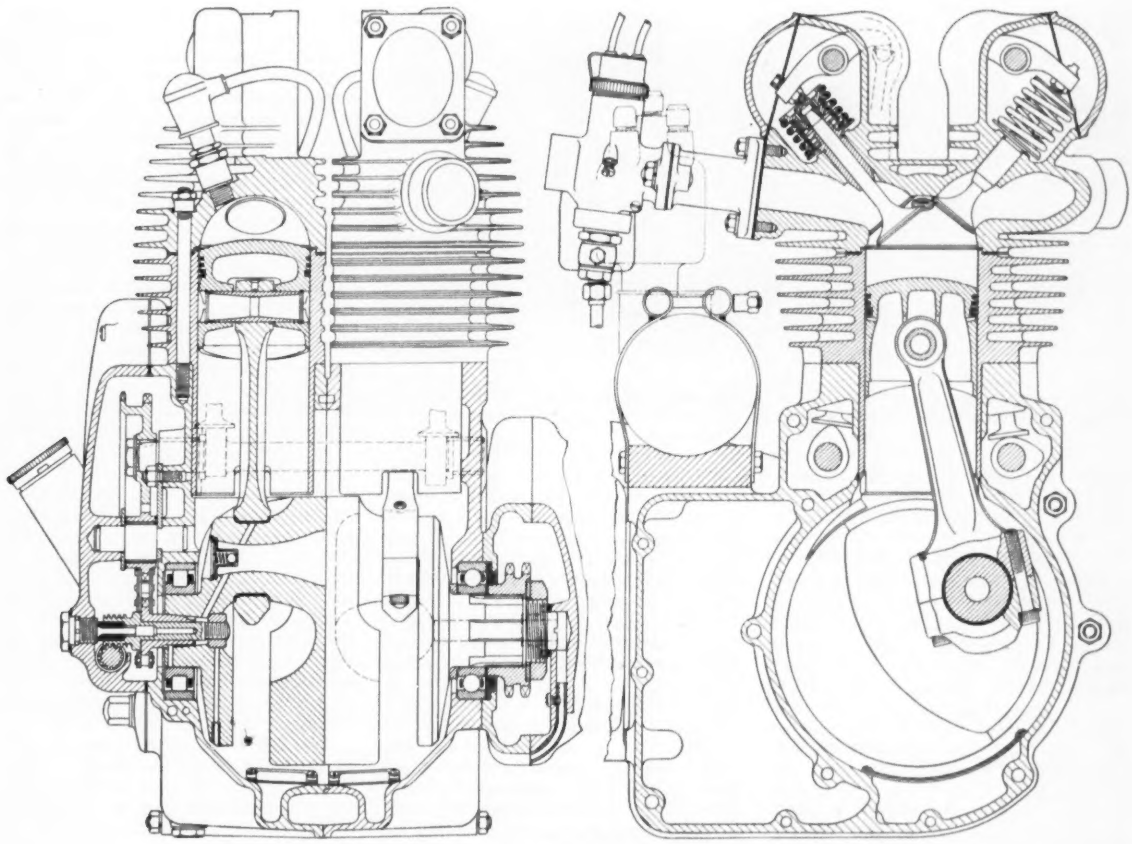
In fact, most of the objections to both front wheel drives and rear engined cars are concerned with difficulties in transmission design. However, discussion of these is outside the scope of this article. That they can be overcome is evident from the success of cars of this type produced on the continent.

The layout problems associated with the rear engine arrangement are much the same as those already outlined in connection with front wheel drive. Perhaps the main difference is that, from the point of view of accessibility, the engine must be behind the wheels instead of in front. Transmission problems are much the same, except that they are simplified by the fact that the wheels are not steered, and this makes easier the accommodation of a horizontally opposed engine. Although, in most rear engined cars, a vertical power unit is better because it is more accessible, the short overall length that may be obtained with a horizontal engine is most desirable because it helps to keep the centre of gravity of the vehicle further forward. This is a critical consideration in cars of this type because of their oversteering tendency.

Adverse weight distribution might be offset by employing an underfloor engine installation. A glance at the illustrations showing the Dyna Panhard twelve cylinder horizontally opposed air-cooled engine will show the possibilities of such an arrangement. With two horizontally opposed cylinders, the overall depth of the engine need not be more than nine inches, although it may be difficult to accommodate the flywheel in this space. The need to position the auxiliaries at the front and rear might lead to difficulties so far as overall length is concerned, particularly if an air-cooled engine were employed. In a water-cooled unit a thermo-syphon cooling system with an uncon-



The Renault gearbox and final drive unit is 16¾ in long so that the total length of the engine and transmission is 38½ in

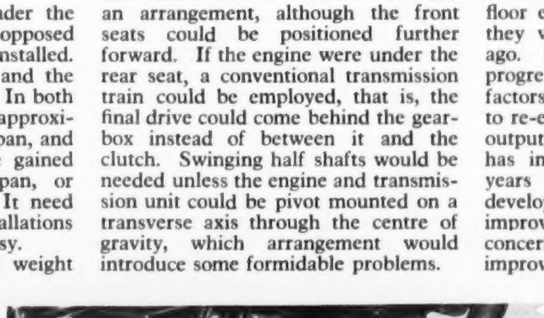


The oil tank is cast integrally with the crankcase of the 692 c.c. Royal Enfield Meteor engine

ventional shape radiator block might be suitable.

There are two positions under the floor where a horizontally opposed twin cylinder engine might be installed. One is under the rear seats and the other is under the front seats. In both places the floor can be raised approximately to the level of the seat pan, and for servicing, access could be gained either by hinging up the pan, or through a removable panel. It need hardly be added that such installations would almost certainly be noisy.

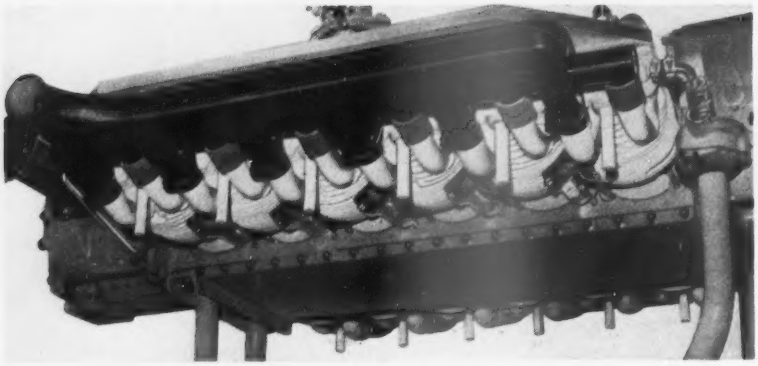
Apart from the improved weight distribution there are a number of advantages to be gained with an underfloor installation. One of the principal objections to the rear engined car, the lack of luggage space, would be overcome, and the weight distribution could be even better than in a normal layout. Moreover, with the engine under the front seat, it might be possible to eliminate the additional universal and sliding joints necessary in both front or rear engined cars and, in addition, the control problem would be eased. However, in very small cars the rear passengers' feet are usually accommodated under the front seats, and this would be difficult with such an arrangement, although the front seats could be positioned further forward. If the engine were under the rear seat, a conventional transmission train could be employed, that is, the final drive could come behind the gearbox instead of between it and the clutch. Swinging half shafts would be needed unless the engine and transmission unit could be pivot mounted on a transverse axis through the centre of gravity, which arrangement would introduce some formidable problems.



The oil filler tube may be seen projecting from between the gearbox and engine timing cover of the Royal Enfield Meteor

It is usual to assume that if unconventional arrangements such as underfloor engine layouts were practicable, they would have been adopted years ago. However, conditions change as progress is made, and there are two factors that make it worth while now to re-examine the problem. One is that output in terms of horsepower per litre has increased considerably of recent years and this, together with other developments, has led to a marked improvement so far as compactness is concerned. The second is that recent improvements in chassisless construction have led to a general clearance of obstruction from the underfloor space. It may well be that, despite these changes, there is still not enough room for the engine and transmission except at the extreme front or rear. Nevertheless, the idea should not be dismissed without careful study.

What effect a centrally positioned engine would have on road holding and ride is difficult to predict. Directional stability would most likely be adversely affected, although control ought to be lighter. It would appear that the effective ratio of sprung to unsprung mass



On the Dyna Panhard 12 cylinder, horizontally opposed, air-cooled engine, the inlet and exhaust manifolds are more or less in line with the cylinder axes, and the torsion bar valve springs extend downwards

would be lower than if the engine were directly over the wheels. This might be detrimental to ride.

Air or water cooled

There is little doubt that the air-cooled installation, because no radiator is needed, is the lighter and more compact arrangement. This is probably the main reason why it has been adopted in the Citroen 2CV, Volkeswagen, Dyna Panhard with the horizontally opposed twin cylinder engine, and the Nardi, in which the Dyna Panhard engine is used. The principal disadvantage of the air-cooled engine is that it is generally noisy. Whether or not this difficulty can be overcome by insulating the engine compartment remains to be proved, but the adoption of elaborate measures is hardly likely to be practicable in cars that are to be produced inexpensively.

So far as cost is concerned, there would appear to be little to choose between the two cooling systems. Air cooling eliminates the water pump and drive but introduces a fan, ducts and baffles. Water jackets are not employed, but with cast fins the overall cost is probably about the same, and if the fins are machined to obtain close spacing it is certainly more. The radiator, water pipes and thermostat are not required, but air-cooled engines run hotter so that a dry sump lubrication system must be employed, and this usually involves installing a separate oil tank and pipe connections. Such an arrangement costs more than a simple pressed steel sump. Moreover, a scavenge oil pump must be added and possibly an oil cooler as well.

Unless siamesed cylinders are employed in a water-cooled engine, the minimum distance between the walls of adjacent cylinders is about $\frac{3}{32}$ in. This is approximately the same as that obtainable between air-cooled cylinders, although with air-cooled engines the tendency is for the space requirement to be slightly larger. Siamesed cylinders are employed successfully by some manufacturers, but many prefer to avoid this arrangement because of the distortion problems that may arise.

General design

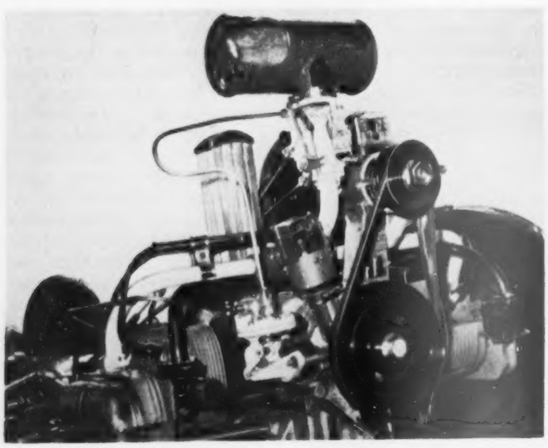
It has already been stated that once the general layout has been decided further gains can only be of much significance if they are the cumulative results of a number of small improvements. This is because nearly all engines in the quantity produced cars made in this country are almost as compact as they can be made without introducing undesirable features. In fact, almost every change that might be made, can only be effected if increased costs are acceptable.

Nevertheless, it must be pointed out once again that costs should be assessed with regard to the whole installation, and not only from the restricted point of view of detail design. Moreover, there are occasions when some expensive features must be incorporated in the engine in order to make practicable a more suitable overall vehicle layout, which otherwise could not possibly be adopted. In other words, good design is nearly always a compromise.

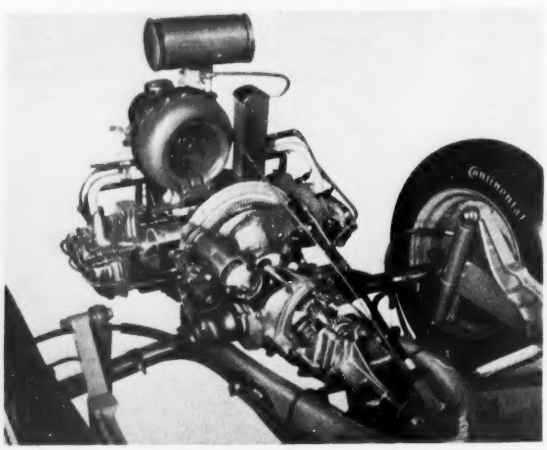
Where compactness is of prime importance it is suggested that the best procedure would be to prepare a layout, and arrange all the details almost regardless of cost considerations. Then, having achieved the greatest possible degree of compactness, decisions must be taken as to what compromises are to be introduced to make the unit a sound commercial proposition. Only in this way is it likely that appreciable saving in space will be effected.

The first stage in the design process is to decide what the swept volume shall be. This will be determined from the power needed and the b.h.p./litre to be expected. The b.h.p./litre will depend to a large extent on the valve gear and combustion chamber arrangement, and whether a two stroke or four stroke engine is contemplated.

In small engines the number of cylinders and their layout will be determined by three main considerations. The shape of the space available will dictate whether a vertical or horizontal engine is to be employed. A large number of cylinders adds to the cost and, because of the need to accommodate the cylinder walls and

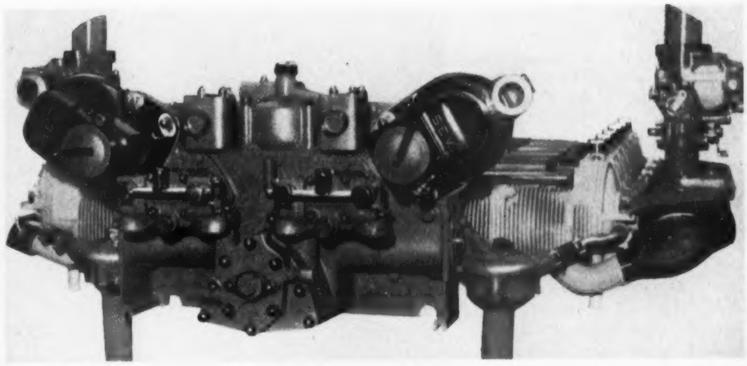


On the rear engined Volkeswagen the blower, as well as supplying cooling air to the cylinders, also serves an oil cooler which may be seen here, since the ducting has been removed



This view from the front of a sectioned Volkeswagen engine shows the transmission arrangement and the way in which the starter motor is mounted to the rear of the flywheel

cooling spaces around them, the fewer the cylinders the shorter the engine. On the other hand, smoother running can be obtained with a multi-cylinder unit. If an exceptionally high performance is required, it is advisable to use smaller cylinders in order to reduce the length of flame travel in the combustion chambers and thereby offset any tendency to detonation.



All the auxiliaries are mounted on the ends of the Dyna Panhard 12 cylinder engine. In certain installations this could make them readily accessible

Length

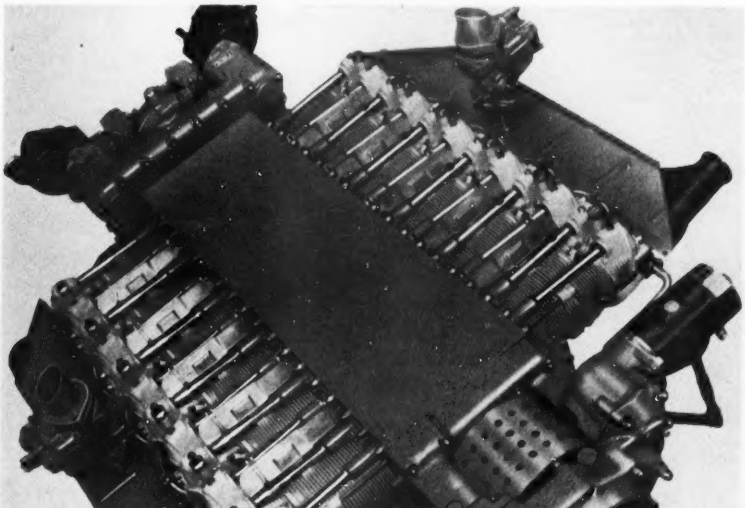
There are three features that have a major influence on the overall length of an engine. One is the spacing of the cylinder axes, another is the journal bearing lengths, and the third is the valve gear arrangement. Cylinder spacing is largely dependent on the bore:stroke ratio adopted, while bearing length is affected by bearing diameter and engine speed. The object, therefore, is to obtain a good all-round compromise between these two features.

Very close cylinder spacing has been obtained with wet liners inserted from below into an integral cylinder block and head unit. The upper end of the liners may be spigoted into recesses in the head and rubber sealing rings are fitted round the flanged lower ends. This layout eliminates dimensional restrictions imposed by practical considerations concerning coring and casting, but it is expensive.

Since swept volume varies as the square of the bore and directly with the stroke, a reduction in the length of the stroke only calls for a relatively small increase in the bore to maintain the same swept volume. On the other hand, bearing loads, and therefore bearing lengths, decrease approximately in direct proportion to the stroke reduction. Thus, if two curves, one showing the variation of bore and the other of bearing length with stroke, are superimposed, the point where they cross will indicate the optimum bore:stroke ratio, so far as engine length is concerned. To one side of this point the engine will be longer than is necessary to accommodate the bearings; on the other side it will again be longer than it need be, but this time the bearings will be so long as to lead to wider cylinder spacing than is necessary.

Another variable, namely engine speed, must also be taken into account. Power output, within limitations imposed by breathing requirements, varies directly with r.p.m. On the other hand, bearing loads and therefore lengths vary as the square of the r.p.m. Thus, a high engine speed is required to obtain the greatest possible power output per litre but, where compactness is essential, a positive limit is reached when the bearing length needed to keep the bearing pressure down to the permissible level exceeds that which may be accommodated within the length of the cylinder block. Under certain conditions, valve port area may also be decisive, since this area must be increased in direct proportion to engine speed; this aspect will be discussed later.

It is well known that lead bronze bearings are capable of withstanding appreciably higher pressures and can therefore be smaller than white metal ones. From this it is reasonable to assume that in a compact unit the bearings will be of lead bronze. There is a limit below which the length of a bearing cannot be reduced; this limit is imposed by difficulties associated with the distribution of oil over the rubbing surfaces and its retention in the bearing. The limitation is even more marked when lead bronze is used, because greater radial clearances have to be adopted. Small bearing areas lead to greater mechanical efficiency. Although the increase in efficiency obtained in this way is unlikely to lead to a noticeable economy in engine size, the cumulative effects of a number of measures may be appreciable.



The dynamo is mounted at one end of the Dyna Panhard engine

Minimum bearing diameters are determined largely by crankshaft strength and stiffness requirements. The shorter the crankshaft the smaller the diameter. Shortening the stroke increases the stiffness by bringing the crank pins and main journals close together. If the bearing area is kept constant, the frictional torque at a given rotational velocity decreases as the diameter is reduced, not only because of the reduced length of torque arm but also because of the reduced rubbing velocity. Therefore, from the point of view of mechanical efficiency, it is desirable to keep the diameter as small as possible.

Nitralloy steel crankshafts probably have the highest fatigue resistance of all the materials currently in use and they may therefore be made smaller. The nitrogen hardening process gives

excellent wear resistance, moreover, it increases the fatigue resistance by as much as 20 per cent. This material is particularly suitable for applications where high bearing pressures are likely to be experienced, and where lead bronze bearings are to be employed.

The Dyna Panhard, 745 c.c. engine installed in the vehicle which was the winner on handicap and in its own class in this year's race at Le Mans

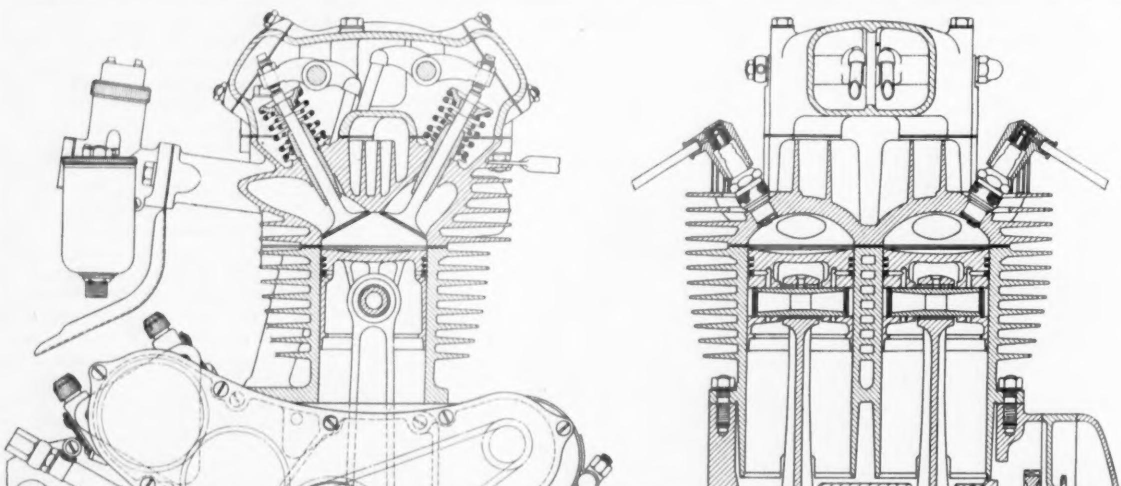
Roller main journal bearings may be used to reduce overall length, but they are both expensive and noisy. An outstanding example of length reduction was to be found in an early Maybach design in which the rollers ran directly on circular crank webs. In this way, the bearings and webs were made coincident; thus the saving per pair of cylinders, so far as length was concerned, was equivalent to the thickness of two crankwebs. However, peripheral speeds are likely to be high with this arrangement.

A number of combinations of roller, ball and plain bearings are employed on motor cycle engines. On single cylinder units, a single ball bearing is often employed at the timing gear end of the crankshaft to take the thrust as well as the journal loading, and a roller bearing is fitted at the drive end.

In twin cylinder engines, it is usual to have only two main journals, one at each end of the shaft, and the flywheel is either integral with the shaft or bolted on to it between the two cranks. As may be seen from the illustrations showing the B.S.A. and Enfield vertical twin engines, this arrangement leads to a considerable reduction of crankshaft length because the flywheel is accommodated in the space, which in any case cannot be reduced, between the cylinder axes. Cast crankshafts are often employed with this layout.

Where an intermediate bearing is employed, it is usually plain. Roller bearings are sometimes used but, if they have to be threaded over a one-piece crankshaft, their diameter is large and therefore peripheral speeds are high. Moreover, it may also be necessary to fit divided distance rings round the shaft to take up the space between the inner race and the journal, otherwise a built-up crankshaft must be employed. Neither of these arrangements would appear to be ideal.

Roller bearings are sometimes employed in big ends, but the arrangement tends to be rather heavy and it is difficult to keep down the overall diameter. Moreover, if engine speeds are high, trouble may be experienced through the rollers being flung outwards by centrifugal force. Where they are employed, they often bear directly on the crank pin and have an outer race in the big end. One manufacturer employs a three row bearing, the rows being separated by spacer

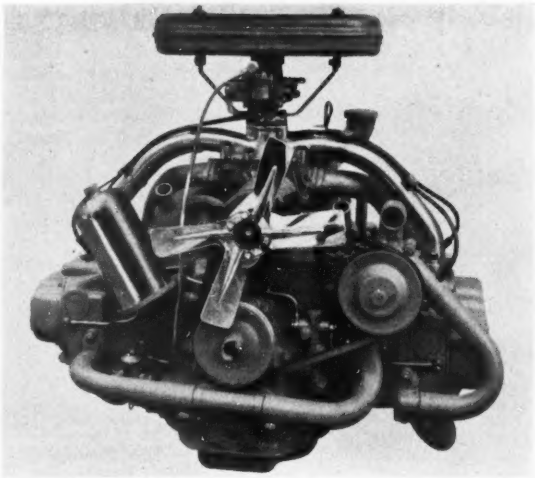


The flywheel of the B.S.A., 646c.c. Golden Flash engine is bolted to the crankshaft and is positioned between the two connecting rods

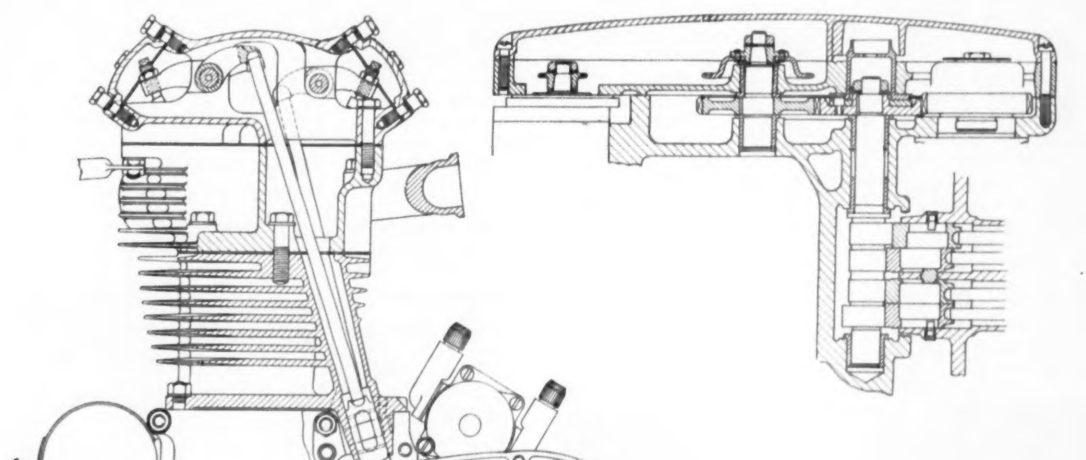
rings. In this way the overall diameter is kept as small as possible. Oil is often fed to the big ends at a low pressure, but main journals are usually lubricated by splash.

On in line and horizontally opposed four cylinder engines in which a four-throw crankshaft is fitted, an intermediate bearing must be employed. Although in some earlier units, there were only two bearings, the loads on modern high speed engines are now too great for such an arrangement to be practicable. The loads on the centre bearing must also be relieved by fully balancing the crankshaft.

Because of the need to accommodate an intermediate bearing, the reduction in length obtained by adopting a flat four design is not so great as might at first be supposed. The offset of the opposing cylinders is increased by the thickness of the crank web as well as by that of two halves of a big end bearing. A slight reduction in length can be obtained by offsetting the connecting rods in each pair of opposed cylinders towards each other, that is, to one side of the centre of their big end bearings. However, the amount that can be gained in this way is only about $\frac{3}{32}$ in per connecting rod. The Jowett Javelin and the Hotchkiss Gregoire engines, shown in the accompanying illustrations, are examples of designs of the flat four type. In the case of the Jowett Javelin, the bore and stroke are 72·5 mm and 90 mm respectively. The overall dimensions of the unit are as follows: width 32½ in over the rocker cover retainer bolt heads; height, 19 in to the top of the water outlet, which is integral with the timing cover, or 26¼ in to the thermostat housing above the fan drive spindle; length, 16⅛ in from the rear of the crankshaft to the front of the starter dogs. Study of these two illustrations will show in what ways the layout is more expensive than a conventional four in line cylinder arrangement. One feature that is worthy of special mention is the Jowett cooling system which is particularly well planned. The coolant flow is from the radiator bottom tank, through separate pipes to the rear of each cylinder block, which it enters near the top and towards the head. These pipes are short because the radiator is behind the engine. After circulating through the heads and blocks, the water passes out through ports high up in the front of each block to passages cast on each



A hot-spot is incorporated below the riser pipe for the single carburettor of the Hotchkiss Gregoire engine

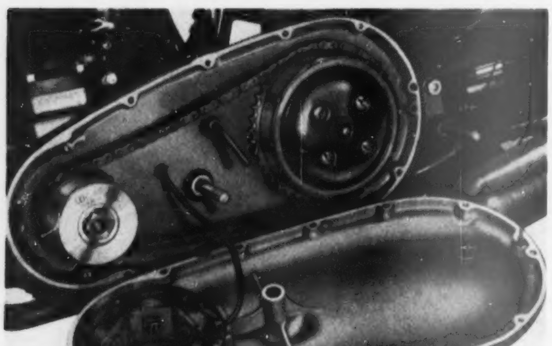


In the B.S.A. Golden Flash engine, which has a bore and stroke of 70 mm and 84 mm respectively, the tappets are carried in pairs to fit in the space available between the two cylinders

The twin downdraught carburettor arrangement of the Jowett Javelin is good from the point of view of accessibility and of induction system efficiency

side of the timing cover. These passages unite at the top in a vertical outlet, on which is a short hose connection to the water pump. A full description of this engine was given in the October 1948 issue of *Automobile Engineer*.

Two downdraught carburettors are employed on the Jowett, one on each cylinder head unit, but on the Hotchkiss a single downdraught carburettor is mounted above the centre of the engine. It is bolted to a rather long arched inlet pipe under which is a hot-spot casting. The exhaust from one side of the engine is carried through another arched pipe and the hot-spot casting, to join the exhaust pipe from the cylinders on the other side. The hot-spot doubtless is of some advantage for running under very cold conditions, but the arrangement is somewhat complicated and it interferes to a certain extent with accessibility.



On the Triumph Speed Twin, an A.C. generator, driven directly by the crankshaft, is mounted in the primary chain cover

There are ways of eliminating the offset of cylinder axes in horizontally opposed engines. However, not only are they generally expensive, but other problems are introduced, notably in connection with lubrication. A method that was adopted many years ago in an A.B.C. horizontally opposed twin cylinder engine was to use a three-throw crank. The centre crank pin was about twice the width of the two side ones and it carried the big end of the connecting rod for one cylinder, while the two outer pins carried two more connecting rods. The small ends of both of these rods were mounted on the gudgeon pin in the piston of the other cylinder. If this arrangement were used in a unit of high power output, the narrow bearings presumably would give rise to some lubrication difficulties, moreover, it is somewhat complicated.

Another method is to use a single throw crankshaft to carry the two connecting rods either with one forked big end, between the prongs of which is the plain big end of the other rod, or with the two big ends side by side on the crank pin. Much the same objections and difficulties arise with these arrangements as with the A.B.C. method but, in addition, some fairly heavy out of balance forces are experienced.

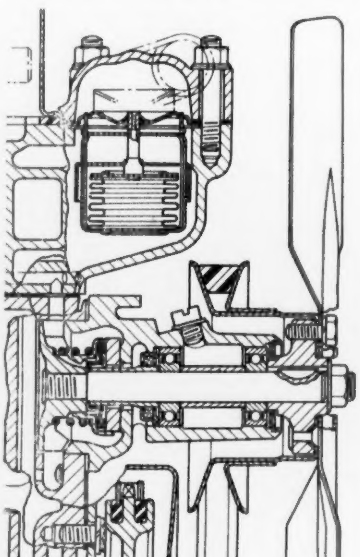
Valve gear design, as has already been stated, may also have an effect on the overall length of the engine. Undoubtedly the simplest and least expensive arrangement is the one in which all the valves are in one straight line. With this arrangement a single rocker shaft may be employed, and all the push rods, tappets and their housings are also in line, so that machining etc. is simplified. In some engines, the valve seats overlap the cylinder bore by a small amount, but there is a limit to the advantage that can be gained by this measure. This limit is imposed not only by the considerations of gas flow and combustion chamber shape, but also because of the need to maintain a reasonable width of gasket between the cylinder bores. Larger valves, leading to improved volumetric efficiency, may be used with spherical or penthouse type combustion chambers. However, this arrangement complicates the valve operating mechanism.

An inexpensive way of accommodating valves larger than may be seated within the limits imposed by the bore size is to use an F-head layout. With this, it is usual to have overhead inlet valves and side exhausts. In this way, the inlet ports are directed conveniently upwards to the carburettor and the exhaust ports downwards to the exhaust pipe. The F-head arrangement has the disadvantage that it is far from ideal from the point of view of combustion efficiency.

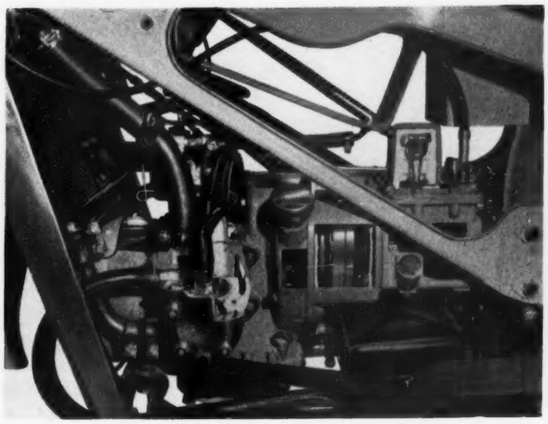
There are other features which affect the length of an engine. For instance, a small increase in flywheel diameter will enable a marked reduction to be made in its thickness. The length occupied by the water pump can be reduced, as in the Austin A30, by spigoting it into the front of the cylinder block. In this engine a semi-shrouded rotor is employed, with the shroud at the rear, so that there is no need for a wall to close the back of the pump casting.

Height

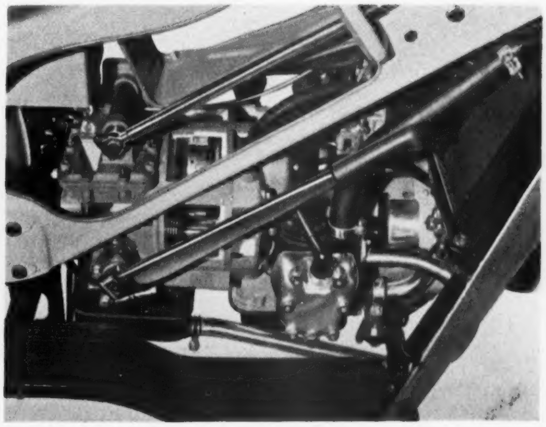
Among the various factors that affect the height of a vertical engine, the bore:stroke ratio is important. This is because an increase in bore and consequent reduction in stroke enables a



The Austin A30 water pump is spigoted into the front of the cylinder block



The Velocette water-cooled, horizontally opposed, twin cylinder engine is particularly silent running



A pull-up lever, instead of the usual kick starter, is used on the Velocette

reduction to be made in the connecting rod length since the ratio connecting rod length:stroke will remain substantially constant. For a given compression ratio, the height of the compression space above the piston at top dead centre is also reduced. The limit to the amount by which the stroke may be reduced is often determined by the clearance, at bottom dead centre, between the piston skirt and the balance weights on the crankshaft. The length of the piston skirt below the bottom compression ring should be no less than ¾ of the bore, and it is better if ⅞ of the bore can be allowed.

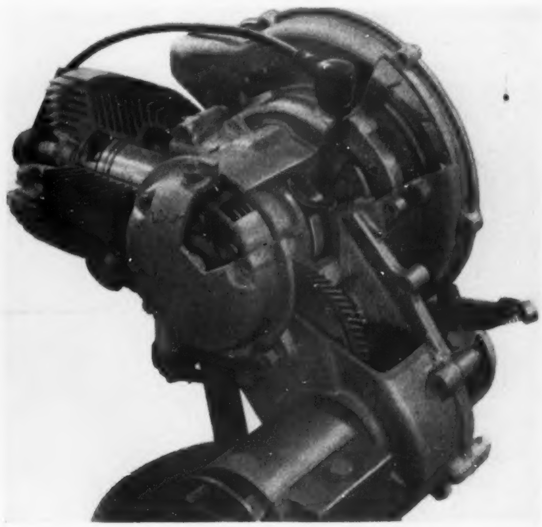
Valve lift is determined largely by considerations of breathing efficiency. Often it is not realized that reducing the valve lift has a twofold effect. Firstly, the restriction of the aperture prevents the attainment of the optimum volumetric efficiency, and secondly it lowers the orifice coefficient and further reduces the flow of gas into the cylinder.

In overhead valve engines it is possible to arrange the pivot axis of the rockers so that the highest points of the whole mechanism are the tops of the tappet adjusting screws. This is current practice on most engines. Various forms of valve spring, such as the hairpin and torsion bar types, may be used to reduce the height of the engine relative to that which would sometimes be necessary to house the relatively long coil springs. However, both these devices were originally developed for air-cooled engines because springs of these types may be positioned well out in the airstream so that they will not lose their temper. Coil springs are generally well recessed into the top of water-cooled cylinder heads, and in any case, it is often possible to reduce the spring length by increasing its overall diameter.

The height of the cylinder head is determined by the valve ports and the valve guides. A compact arrangement may be obtained with a hemispherical or penthouse type of combustion chamber. The advantages of this layout are that the valve ports are not so curved, and the valve guides are inclined at a fairly large angle from the vertical so that they may be fairly long without occupying a large amount of vertical space.

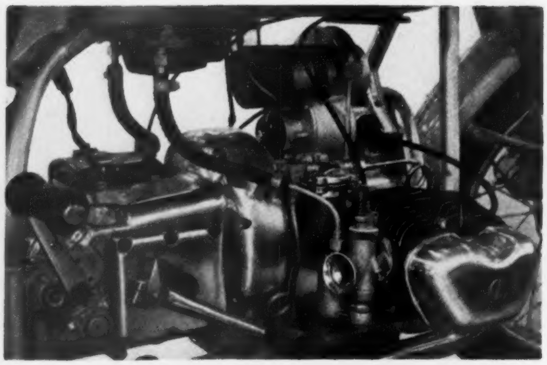
An appreciable saving in height can, of course, be effected by using side valves. The modern trend is towards the almost universal adoption of overhead designs, but this is partly because of the inaccessibility of side valve gear on vertical in line engines. This objection does not apply to horizontally opposed engines when the tappet chest is at the top. It is still open to objection because of relatively poor combustion efficiency, but the flexibility of the side valve engine is a good feature on very small cars which may be somewhat underpowered. Moreover, it is relatively inexpensive.

On the Triumph Terrier, the rectifier and coil are mounted under the saddle, and an A.C. generator is housed in the primary chain cover where it runs in oilmist



The Douglas Vespa has a blower on the right-hand side for supplying cooling air to the cylinder, and the gearbox casing is formed by an extension of the crankcase casting

The bore and stroke of the Corgi are both 50 mm, and the overall depth of the engine is only 7¼ in



The rockers are readily accessible on the Douglas horizontally opposed air-cooled, twin cylinder engine

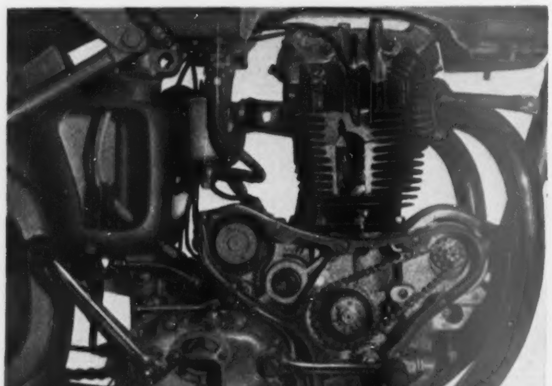
The two-stroke engine has much to commend it from the points of view both of compactness and of simplicity. The principal disadvantages of this system are a tendency to four-stroking and stalling at low speeds, a high brake specific fuel consumption under light load, and its noise. The brake specific fuel consumption can be made comparable with that of a four-stroke engine by employing petrol injection. Most of the injection equipment currently in use is expensive, but because of the low pressures involved in equipment for this type of engine it might be practicable to use a bellows type of injection pump instead of the conventional barrel and plunger arrangement with its close manufacturing tolerances. The output per litre of a modern two-stroke engine compares more than favourably with that of four-stroke units. For instance, the three cylinder Hanomag engine, which has a swept volume of 697 c.c., develops 40 b.h.p./litre; the two cylinder 594 c.c. Gutbrod develops 38·6 b.h.p./litre, and a figure of 38·0 is obtained with the three cylinder 896 c.c. Auto Union D.K.W.

At the base of the engine, economy in vertical space is often obtained by using a blister type sump, that is, one with its sides and front end extended outwards. This type of sump is usually cast, because of the difficulty of avoiding leaks with the alternative construction in which it is pressed in two pieces and welded together at a joint in a vertical plane. Whichever method of manufacture is employed, the cost is appreciably higher than that of a simple dished steel pressing. Another method of economizing in height is to cast the sump integrally on the side, front of the crankcase, or on the rear as in the Enfield engine illustrated. This is a dry sump system, so both a scavenge and pressure oil pump must be employed. Nevertheless, it is not such an expensive layout as those in which a separate tank is used. Moreover, there is no danger of failure of the lubrication system as a result of broken pipes or connections.

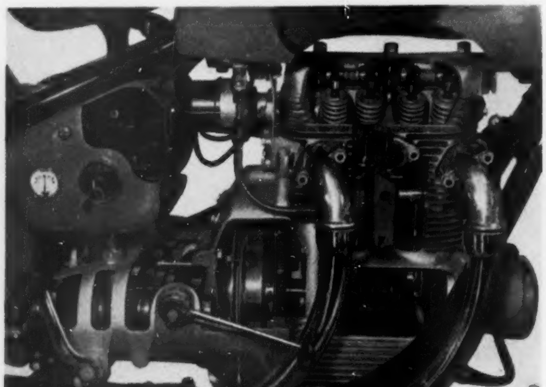
In horizontally opposed engines the height can be reduced by arranging the inlet and exhaust manifolds in such a manner that they are in line with the cylinder axes, as on the Dyna Panhard 12 cylinder engine illustrated. This restricts the accessibility to the valve gear, and it is no doubt one reason for the incorporation of torsion bar valve springs in that unit.

Further reduction in height on horizontally opposed engines, as well as reduction in width on vertical units, can only be gained by positioning the accessories at the front and rear. On both the 12 cylinder Dyna Panhard and the Citroen 2CV the dynamo is driven directly by the front end of the crankshaft. The cooling fan on the Citroen is mounted in front of the dynamo and serves an oil cooler as well as the air-cooled cylinders. The starter is not so easily accommodated, but in some cases it might be possible to construct the flywheel in the form of an internally toothed annulus, to engage the starter motor pinion, thereby bringing the axes of the starter and crankshaft axes closer together. With such an arrangement the starter would, of course, be bolted to the rear face of the flywheel casing.

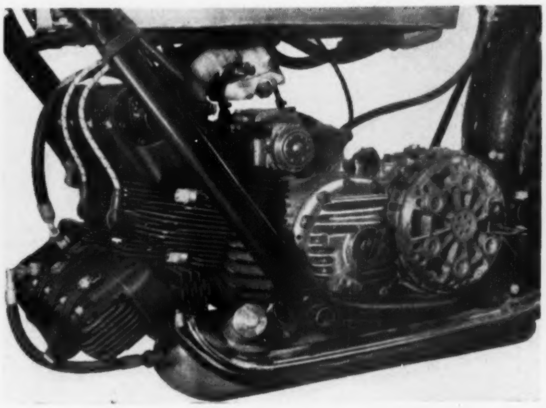
Considerable economies might be obtained by dispensing with the starter motor altogether. Then it would be possible to use a much smaller battery. The crankshaft could be turned either by means of a pull-up lever, such as



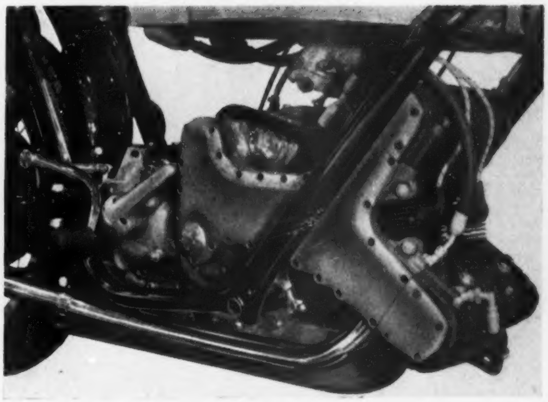
The B.S.A. Golden Flash 646 c.c., twin cylinder engine



The two cylinders of the Sunbeam S7 are siamesed



On the A.J.S. Porcupine, the finning on both the head and the barrel is parallel to the cylinder axes



The rockers of the twin cylinder, A.J.S. Porcupine are in an accessible position

may be seen in the illustration of the flat twin, water-cooled Velocette, by a kick-down type starter, or by an ordinary hand crank.

Motor cycle engines

Motor cycle manufacturers are most concerned with compactness, and for this reason their designs are of interest. In general, the output in terms of b.h.p./litre obtained from a motor cycle engine is very high; this leads to considerable saving in space. There are a number of reasons why such a high specific power output can be obtained from motor cycle engines, one being that induction systems are usually much simpler than on cars, and another is that overhead valves with hemispherical combustion chambers are frequently employed. The incentive towards development on these lines arises from the fact that in this country motor cycle taxation is based on swept volume. However, the lengths to which some motor cycle designers go in order to obtain compactness demonstrates that in this class of vehicle economy of space often is even more important than the cost of the product. The accompanying illustrations show examples of many of the layouts employed.

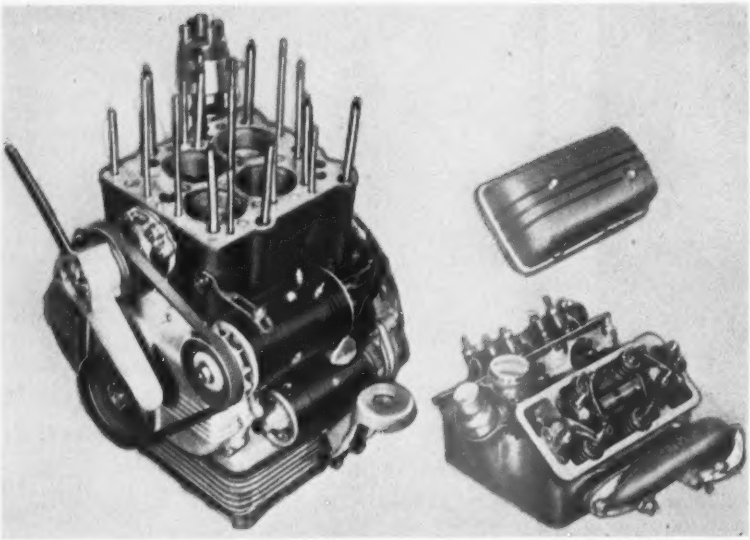
An interesting feature of the 149·3 c.c. Triumph Terrier is its electrical system. A Lucas A.C. generator is totally enclosed by the primary chain cover and is used in conjunction with a rectifier. This generator is of the rotating magnet type, the rotor unit being on one end of the crankshaft. A rotating magnet unit is essential since it has to run in oil splash and mist, so carbon brushes and commutator rings are out of question. The unit is of relatively large diameter and short length so that it may be accommodated in the space available. The stationary part of the generator is bolted to the primary cover in a similar manner to that shown in the illustration of the Triumph Speed Twin generator. Coil ignition is employed and, for emergency starting when the battery is flat, a switch is incorporated in the circuit to direct the current through the rectifier to the coil. In the illustration of the Terrier, the rectifier may be seen with the coil above it under the saddle.

Engines in which the cylinder axis is parallel to the longitudinal axis of the vehicle are of some interest. Among these are the two-stroke Corgi, the overall depth of which is only 7¼ in. This engine has a bore and stroke of 50 mm by 50 mm. The 125 c.c. two-stroke Douglas Vespa, with a bore and stroke of 56·5 mm × 49·8 mm, occupies more space vertically because it is designed to suit an installation having a direct gear drive instead of a chain drive.

A more conventional installation of this type is the A.J.S. Porcupine. This is a horizontal twin engine. The cylinders are side by side, with their axes parallel to the longitudinal axis of the machine. The cylinder heads face forward, so that not only are they well cooled but the overhead valve gear is in an exceptionally accessible position. The fins on the cylinder barrels of the Porcupine, unlike those of the single-cylinder Corgi and Vespa units,

On the Vincent Black Shadow the carburettor for the front cylinder is on the left, while that for the rear is on the right

The valve springs are above the rockers and stepped valve stems are employed on the Black Shadow



One cylinder block is used for the Lancia Appia V-four engine

are arranged longitudinally instead of around the periphery. All three engines have fins arranged longitudinally on their cylinder heads.

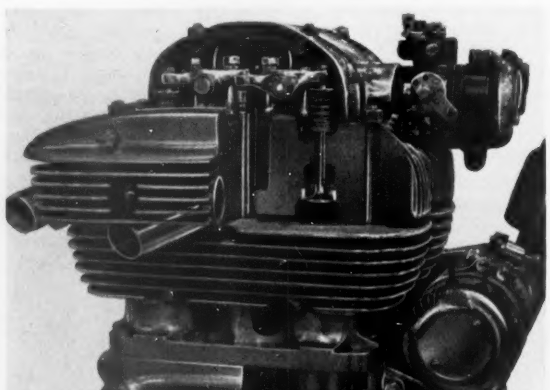
A number of different twin cylinder layouts are employed. The 692 c.c. Royal Enfield Meteor is an example of a twin side-by-side arrangement. It is particularly interesting because the oil tank is cast integrally with the crankcase, which is split vertically along its longitudinal centre line. The highest oil temperature which has been measured on bench tests of this engine is about 180 deg F. Moreover, no trouble has been experienced on long distance races, so the arrangement is evidently satisfactory.

In common with most of the vertical twin engines, this unit has a single-throw crankshaft. Although vertical out-of-balance forces are greater than they would be with a 180 deg crank arrangement, the torque is more even since the cylinders fire on alternate strokes. The bore and stroke of this engine are 70 mm and 90 mm respectively and the overall depth is approximately 20 in. A cast crankshaft is employed with an integral flywheel between the two crank pins. It is carried in a roller bearing on the timing gear side of the crankcase and a ball thrust bearing on the drive side.

Another vertical twin arrangement may be seen in the illustration showing the B.S.A. Golden Flash. This is a 646 c.c. engine with a bore and stroke of 70 mm and 84 mm respectively. Its overall depth is approximately 21 in and its width is 13¾ in. The crankshaft arrangement is much the same as that employed in the Royal Enfield Meteor, but the flywheel is bolted on. With all engines of this type it is desirable, of course, to have a timed breather, since an unduly large amount of oil tends to be lost through an open type breather. That on the Golden Flash opens at 84 deg 25 min after top dead centre and closes 33 deg 34 min after bottom dead centre.

Another example of a two cylinder layout may be seen in the illustration of the 487 c.c. Sunbeam S7. This engine is of particular interest because its layout is in many respects somewhat similar to that of a car unit, although in some details it is not. The two cylinders are in line and the crankshaft is parallel to the longitudinal axis of the motor cycle. The unit is rubber mounted, and the final drive is taken through a shaft that has a needle roller universal joint at the rear and a flexible coupling at the gearbox end. There are, in effect, two flywheels; one is formed by the crankshaft balance

Two crankshafts are employed in the Ariel Square Four and they are geared together 180 deg out of phase



The push rods of the Ariel Square Four are housed in a transverse plane between the front and rear pairs of cylinders

weight between the cylinders and the other, on which is mounted a single dry plate clutch, is at the rear.

The differences between this engine layout and most car units have arisen mainly because of the need to restrict the overall width. For instance, the Lucas 45L generator is driven directly from the front of the crankshaft, and a Lucas A654 distributor is driven from the rear of the overhead camshaft. A ⅜ in pitch chain drive for the camshaft is taken from a gear and sprocket at the back of the engine. The sprocket is on the front of the gear which engages another gear on the rear end of the crankshaft. This arrangement allows a fairly short chain to be used and the reduction to half speed is effected in the gearing instead of by means of the sprocket, so that the width of the space occupied by the chain is reduced to a minimum. An automatic chain tensioner is employed. The cams bear directly on the ends of the valve rockers.

Aluminium cylinder barrels and heads are used in conjunction with slip fit Brivadium liners. The bore and stroke are 2¾ in and 2½ in respectively, and alternative compression ratios of 6·8:1, 6·5:1 or 6·21 are available. A single throw crankshaft is employed. The front journal is carried by a deep groove ball bearing, and at the rear a Glacier white metal shell-type bearing is employed. The connecting rods are of light alloy with Vandervell shell-type big end bearings.

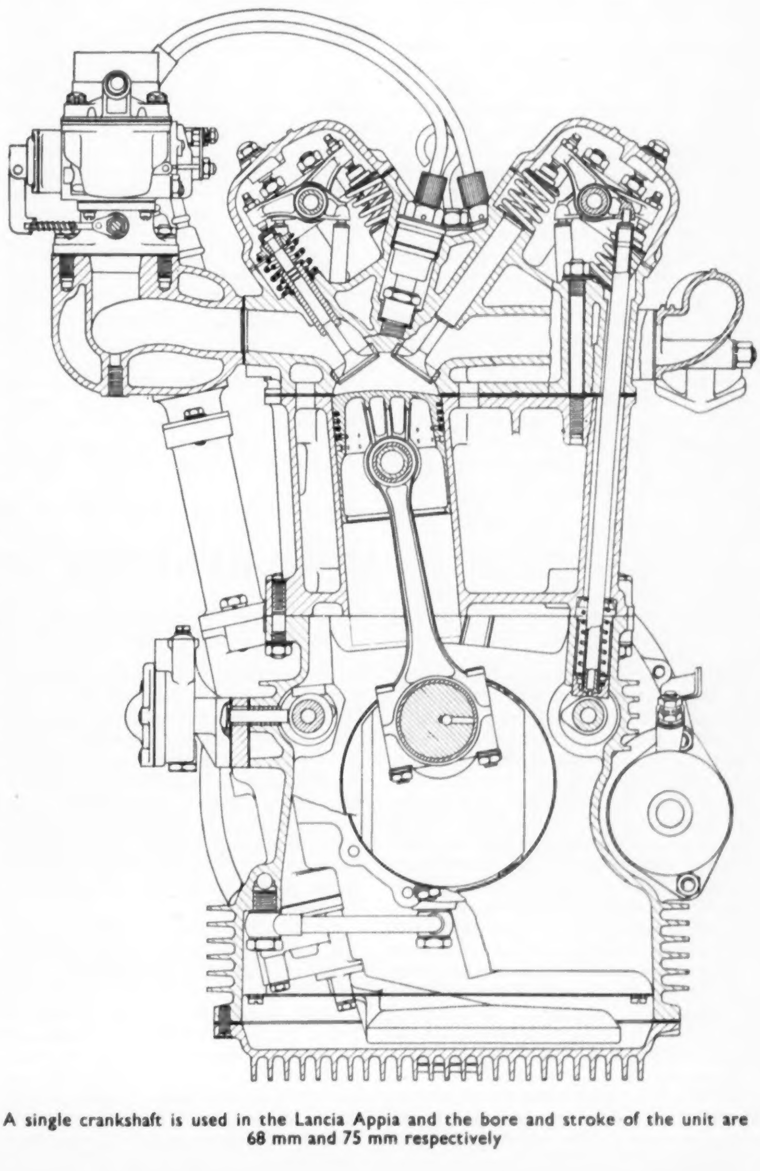
The cylinders, which are integral with the crankcase, are siamesed and the head is also in one piece. Presumably this arrangement would not be practicable if it were not for the high thermal conductivity of aluminium. The manufacturers state that the rear cylinder runs cooler than the front one. This is attributed to deflection by the driver's legs of the airstream on to this cylinder. The crankcase is well finned, and a wet sump has been adopted.

A twin cylinder layout that is commonly used on motor cycles is the 50 deg Vee. This angle is adopted because it gives reasonably even firing together with satisfactory balance. In other words, it is a compromise. In the Vincent Black Shadow, which is of this type, a single throw crankshaft is employed and the cylinder offset of 1½ in is arranged so that the exhaust side of the rear cylinder is further out in the airstream than would be the case if a forked connecting rod arrangement were used to place the cylinders directly in line. Cooling around the exhaust ports is an important feature of all air-cooled engines, and it is for this reason that the exhaust ports face forwards on most motor cycles.

This engine has an 84 mm bore and a 90 mm stroke. It has a swept volume of 998 c.c. and develops 54 b.h.p. at 5,800 r.p.m. The crankcase, cylinders and cylinder heads are all of aluminium alloy and pressed-in liners are fitted. Great pains have been taken to keep the weight of this unit down to a minimum; even the banjo bolts in the lubrication system are made of light alloy.

An unusual feature of this engine, which may be seen in the illustration, is the rocker and valve arrangement. Each rocker is carried on a steel pin in a Y-alloy carrier. This carrier is cylindrical in shape and its centre is machined away so that the rocker may be passed through it, and it is drilled diametrically for the hollow pivot pin of the rocker. Drilled radially in the top of the carrier is another hole in which is engaged the end of a banjo bolt, screwed into the housing, to locate it axially. The banjo is supplied with oil under pressure to lubricate the rocker. Tappet adjustment is effected at the push rod end of the rocker, access to which is gained by removing a cap screwed into the housing. The other end of the rocker is forked and bears on a collar round the stepped valve stem. Two valve guides are employed, the lower one being of larger diameter than the upper. The upper guide is integral with the seat washer for the coil springs, the top ends of which bear against a conventional retainer washer. Although this arrangement places the rockers relatively low, it is doubtful whether much is gained from the point of view of compactness, but in this particular installation it has two advantages. One is that the springs are higher and therefore in a cooler position, and the second is that access for tappet adjustment is good.

Horizontally opposed twin cylinder engines, despite their relatively large overall width, are sometimes used in motor cycles. The Douglas Mark V and 90 Plus models are examples of air-cooled engines of this type. Both



A single crankshaft is used in the Lancia Appia and the bore and stroke of the unit are 68 mm and 75 mm respectively

have a bore and stroke of 60·8 mm by 60 mm and a swept volume of 348 c.c. The Mark V has a compression ratio of 7·25:1 and that of the 90 Plus is 8·25:1. For racing, a compression ratio of 9·4:1 has been used. A built-up, two-throw crankshaft is employed, and the cylinder axes are offset from one another by only $\frac{3}{8}$ in. An overhead valve arrangement is used and it gives the engine a good peak performance; moreover the rocker gear is very accessible. The overall width of the engine is $24\frac{5}{8}$ in and its length, without the gearbox fitted, is $19\frac{1}{32}$ in. The height is $17\frac{1}{2}$ in, but this is in no way restricted by the installation. In fact, in order to fill the vertical space in the frame, the engine has almost certainly been made higher than would otherwise have been necessary.

The horizontally opposed twin cylinder Velocette, 192 c.c. L.E. model is water-cooled. Its bore and stroke are 50 mm and 49 mm respectively. The offset of the cylinder axes from one another is only $\frac{3}{8}$ in, because short roller bearings are used in the big ends. The main reason for adopting water cooling was to produce a quieter engine than was possible with air cooling. Two radiator blocks are employed, one for each cylinder, and they are separated by an air box for the carburettor intake and an Amal air filter. Each radiator block is 4 in wide by $6\frac{1}{2}$ in deep by $1\frac{7}{8}$ in thick.

A pull-up hand lever is incorporated for starting. One pull of the lever turns the crankshaft one revolution. This lever is mounted on one end of a transverse spindle. At the other end of the spindle is another short lever on which bears the end of a similar short lever at the rear of a longitudinally positioned spindle. The action of the short levers may be likened to that of a pair of one-toothed bevel gears meshing together. Since there is a return spring round the longitudinal spindle the hardened faces at the ends of the short levers are kept constantly in contact so that this simple arrangement is all that is needed. At the front end of the longitudinal spindle there is a toothed quadrant that engages one of the gears of a 3:1 reduction train interposed between the crankshaft and the two plate dry clutch.

The Ariel Square Four is of interest, although it is hardly suitable for a motor car, partly because it would be difficult to provide adequate cooling in such an installation and partly because it is rather expensive for the type of car for which it is likely to be required. The overall dimensions are approximately $17\frac{1}{2}$ in long by 15 in wide by 19 in high. It is an overhead valve engine with a bore and stroke of 65 mm × 75 mm respectively and a swept volume of 997 c.c. The cylinder block is cast in one piece, and has a more or less square plan form.

The two crankshafts are geared together and are positioned transversely one under each pair of cylinders. They are one-piece forgings and the flywheel and balance weights are bolted on between the crank throws, which are at 180 deg. The two crankshafts are phased so that they rotate at 180 deg to one another.

A chain drive serves the camshaft which is mounted transversely in the crankcase between the front and rear pairs of cylinders. The push rods are positioned in line in a transverse plane vertically above the camshaft. A conventional rocker arrangement has been adopted, one shaft serving the front pair of cylinders and the other the rear. The four inlet ports are joined at the centre of the engine to form a cruciform arrangement into which the carburetted mixture is fed from above. This gives an even distribution to all the cylinders. Four separate exhaust pipes are used to obviate the difficulties that would be experienced in an engine of this type if two cylinders were served by a common pipe.

It is interesting to compare an Ariel Square Four with the water-cooled Lancia Appia engine. The two units are similar in some respects, but in the Lancia a single, two-bearing crankshaft is employed, and the cylinders are arranged in Vee form, the included angle being 10 deg 14 min. The firing order is 1, 3, 4, 2 so the crank pins of numbers 3 and 4 cylinders are at 180 deg, as are those of numbers 1 and 2. In order to obtain regular firing intervals, the crankshaft is in effect twisted 10 deg 14 min between the crank pins for numbers 2 and 3 cylinders. With this arrangement the length of the cylinder block is only $9\frac{1}{2}$ in and its width 9 in. However, the overall width of the engine is approximately 18 in measured over the inlet and exhaust manifolds on each side of the cylinder head. The bore and stroke are 68 mm and 75 mm respectively, and the swept volume is 1,090 c.c. The compression ratio is 7·4 : 1.

Because a single crankshaft has been employed it has been necessary, of course, to offset the cylinders longitudinally and operate the push rods for the overhead valves by two camshafts, one on each side of the engine. Whereas on the Ariel Square Four the valves are vertical above their flat topped combustion chambers, on the Appia, hemispherical combustion chambers are employed with inclined valves. The general arrangement of the valve gear may be seen from the illustrations.

CABMA REGISTER 1953

An Important New Reference Work

PUBLISHED jointly by Kelly's Directories Limited and Iliffe and Sons Limited, Dorset House, Stamford Street, London, S.E.1, for the Canadian Association of British Manufacturers and Agencies—managers of the British Trade Centres in Toronto, Vancouver and Montreal, the CABMA Register 1953 provides for the first time a comprehensive record of more than 4,500 British manufacturers, their products and their Canadian representatives. The volume is sectionalized and divided for easy cross reference by index guide cards with reinforced tabs.

The first section, the Buyers' Guide, is an alphabetical list of British products for Canada, with the names of the British and Canadian suppliers classified under the product headings. A directory of British manufacturers and distributors is in a separate section. In this section the British firms are listed first with full details, including their distribution arrangements in Canada; a list of Canadian suppliers follows. Two further sections enable the products and sources of supply to be identified from proprietary names and trade marks respectively. To ensure that the Register can be readily used throughout the whole of Canada, there is a complete French-English glossary of the product headings in the Buyers' Guide.

In the preparation of the register, full co-operation has been received from, among other organizations: the Dollar-Sterling Trade Council in Canada; the Dollar Exports Council in the United Kingdom, the Federation of British Industries and the Association of British Chambers of Commerce. With the help of the various organizations, the widest possible survey was made of British firms who export their products to Canada. The publishers then approached all the firms individually, and from the information they supplied and vouched for, the Register has been compiled. Every effort was made to secure the co-operation of each exporting firm concerned to ensure that the Register should reach a high standard of accuracy and completeness.

The volume has been warmly welcomed in official circles in both this country and Canada. It will without doubt become the standard reference work for buyers and importers, and will be in daily use by everyone requiring authoritative information about British goods available to the Canadian market, and it has already been referred to as a constructive step towards the expansion of British trade in Canada. The volume is bound in full cloth and is priced at 42s. 0d. (44s. 0d. by post).

HEAT RADIATION SUPPRESSION

The Bonding of Refractory Coatings to Metal Components that are Subjected to Elevated Temperatures

A RESEARCH report, which deals with the effectiveness, strength and methods of bonding of refractory materials to metals, has recently been issued by the Fulmer Research Institute. It is *Special Report No.* 1, entitled *Radiation Suppressing Coatings for Metals at Elevated Temperatures.* Although the work was done primarily for application to gas turbines, the methods advocated might also be used to reduce radiation to or from metal components for reciprocating type internal combustion engines, notably in exhaust systems.

The object was to develop a refractory coating of low thermal emissivity, which could be applied to alloys used in gas turbine construction. This coating would also be required to withstand the severe operating conditions, involving thermal shock and vibration, which are encountered in certain parts of these mechanisms. The most obvious application of such coatings is to the internal surface of flame tubes. These tubes, in order that their strength and resistance to oxidation may not be impaired, must not be heated to temperatures higher than 800-900 deg C, although the flame temperature is as much as 1,800-2,000 deg C. For highly luminous flames, such as result from the combustion of marginal fuels, the transfer of heat by radiation is a significant proportion of the total heat transferred from the flame to the surrounding wall. Under these conditions the maximum advantage accrues from a reduction of the thermal absorptivity of the inner surface of the wall.

In the research report, the subject is dealt with in three parts. The first part describes the accurate measurement of the total emissivity of both metal surfaces and refractory substances, and the variation of emissivity with particle size and thickness of coating. Next, investigations are made into methods of bonding refractory coatings to metal surfaces, coating compositions, and into methods of application. Finally, fatigue and other tests are carried out to determine the behaviour of coatings under simulated service conditions.

By Kirchoff's law, the absorptivity of a substance is the same as its emissivity for radiation of the same wavelength. A body that absorbs all the radiation incident on its surface is defined as a *black body,* and one for which the absorptivity, or emissivity, is the same for all wavelengths is termed a *grey body.* The research was based on the assumption that the flame tube wall was a grey body. This is an approximation; for not only is its spectrum unlikely to be the same as that of the flame, even if both were at the same temperature, but also because the radiation spectrum of the flame is unknown.

A sensitive thermopile was used to measure the radiations from coatings applied to one face of an electrically heated strip of Nimonic 75, and the strip was then turned so that measurements could be made of the radiation from the uncoated side. High temperatures were measured with an optical pyrometer, while a fine wire thermocouple was used to check low ones. Curves are given in the report showing the emissivity, at temperatures from 300-800 deg C, and with three different surface finishes, of an 80 per cent Ni, 20 per cent Cr alloy, an 18 per cent Cr, 8 per cent Ni austenitic steel, and a 25 per cent Cr, 20 per cent Ni austenitic steel.

These curves show that surface treatment has a marked influence on emissivity. At temperatures up to 600 deg C or more, a polished surface is most effective for reducing radiation. However, because of oxidation, a short period of overheating results in a permanent and considerable increase in emissivity. Another undesirable characteristic of metal surfaces is that

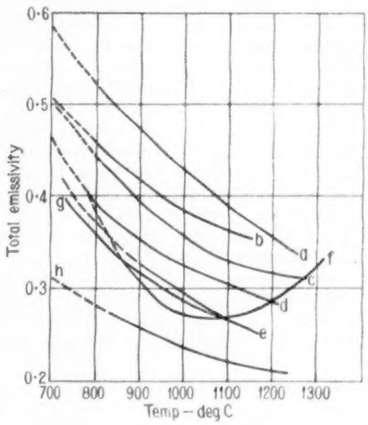


Fig. 1. Total emissivity of various refractory substances at temperatures of 800-1300 deg C

their emissivity increases with temperature. On the other hand, the emissivity of refractive materials decreases as their temperature is increased, Fig. 1.

Several methods of application of low emissivity refractories to the surface were investigated. Electrophorosis was shown to have some promise for the application of thin coatings to an enamelled metal surface. However, neither this process nor bonding with ethyl silicate was successful for applying coatings directly to the metal. The best method was the bonding of the refractory with a proportion of enamel. In this way a composition was formed which softened at the firing temperature of the coating, so that the enamel bonded the particles of refractory material to each other and to the basis metal.

It was found that the size of particles has a marked effect on the efficiency of the coatings. With small particles there is a tendency to spall, or chip, on firing, while large ones, because of radiation from the underlying metal, give a high value of apparent emissivity when applied in thin coatings. In general, the best results were obtained with a refractory that had been ground and passed through a 150 mesh sieve.

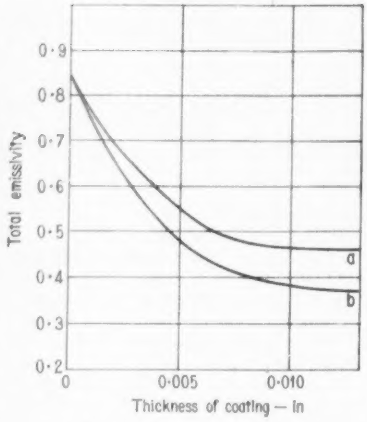
Bonded coatings are porous, so oxidation may take place under them, particularly during the firing process. This adversely affects adhesion in long term service. It is therefore recommended that on most materials the refractory coating should be applied over an undercoating of enamel. The report deals at some length with the methods recommended for the preparation of the surface and for the application of the coatings.

The effects of varying the proportions of enamel and refractory were examined, as was the emissivity of different thicknesses of coating. Fig. 2 shows the type of curve obtained when emissivity is plotted against thickness of coating. Only a relatively small number of different coatings are discussed in the report, but they are the best of a very much greater number that were tested.

To test the adhesion of the coatings under conditions approximating to those obtained in service, a machine was designed by means of which test pieces could be subjected simultaneously to vibration and fairly rapid changes of temperature. Test strips used in the machine were clamped at one end while at the other they were vibrated by an eccentric. A contact was provided at the oscillating end so that a current could be passed through the strip to heat it. The circuit was opened and closed on a 2½ min cycle to vary the temperature between 800 deg C and that of the room. If the strip broke as a result of fatigue, the driving motor was automatically stopped, and an electric clock in the circuit stopped and indicated the time of the failure.

Up to a thickness of about 0·007 in, no significant flaking, of the types of coatings recommended as being satisfactory, occurred before the metal failed as a result of fatigue. With thicker coatings, flaking tended to take place; a ridge or thick area definitely lead to this type of failure. A method of spraying has been developed which minimizes this kind of defect. Vibrating strip tests were also carried out on specimens in which holes had been drilled. Despite the stress concentration effects around these holes, no lack of adhesion was detected.

Thermal shock tests were also carried out. Coatings were applied to test specimens in the form of sheets, 4 in × 4 in × 0·048 in with a central hole 0·585 in diameter. These were alternately exposed to the flame of a gas-air torch directed at the hole, and a blast of cold air. Flaking developed quickly in single coatings. It was caused by the reducing action of the flame. This was demonstrated by the fact that thermal shock arising from electrically heating and air blasting specimens to produce the same heating and cooling rates did not produce flaking. To prevent failure of the coating in a reducing atmosphere, it is necessary to interpose between the porous, low emissivity coating and the basis metal a non-porous layer which will bond both to the underlying metal and to the coating. The National Bureau of Standards coating A 417 has been found suitable for this purpose.



a At 900 deg C b At 1200 deg C

Fig. 2. Variation of total emissivity with thickness of a certain refractory coating

An interesting phenomenon was noted when testing plates under direct flame impingement. Whereas an uncoated plate showed a considerable area of soot deposition, no soot was deposited under identical conditions on the coated plate. Whatever the cause of this may be, its practical importance is obvious; soot deposition, if it occurred in service, would result in the emissivity of the surface approaching that of a black body.

These coatings should be satisfactory for temperatures up to 900 deg C, but they are not recommended for use where the temperature will exceed the firing temperature of the coating. It is not considered that they will afford any protection to the metal against attack by volatile constituents of the fuel, such as vanadium or lead compounds. Indeed, it is possible that the adhesion of the coatings may be adversely affected by such attack. However, there is not yet enough data from practical trials with the coatings for their behaviour with fuels of this type to be fully assessed.

Copies of this report may be obtained for 15 shillings from the Fulmer Research Institute, Stoke Poges, Buckinghamshire.

HIGH SPEED VALVE MOTION

A PAPER entitled "Calculation of High Speed Valve Motion with a Flexible Overhead Linkage," by P. Barkan, is published in an S.A.E. Preprint March 3-5, 1953. In it a method for calculating the high-speed motion of a cam-actuated valve operated by a flexible overhead linkage is described. The actual linkage is divided into several sections; the masses within each section are assumed to be concentrated at a point which is connected to neighbouring masses by massless springs of appropriate rates. In the system, the valve, spring, spring cap and push rod undergo simple reciprocating motion, while the upper rocker arm and the lower rocker lever undergo oscillating rotary motion. In addition, because of the flexibility of the camshaft and the compressibility of the bearing oil film, the camshaft is considered to have reciprocating motion. A dynamically equivalent system is then constructed in which all the parts lie in the same datum line, that is, the valve axis. To make mathematical treatment possible, this multi-mass system in turn is replaced by an equivalent single-mass system.

The mass is subjected to the spring, inertia, linkage compression, friction and gas forces, the sum of which, according to d'Alembert's principle, must be zero. Friction itself is made up of Coulomb friction proportional to load, viscious friction proportional to relative velocity, and viscious friction proportional to absolute velocity. Three differential equations of motion for the mass are developed, that is, for valve opening, valve closing and for the linkage jumping off the cam. Valve motion can be easily determined

The three equations are of the same form, and will have the same general solution. A completely general numerical solution can be obtained by assuming that over a small increment of time, the function may be represented by a linear equation of a type quoted. The problem thus resolves itself into a series of equations, each valid only within a small time increment. The solution depends on the determination, by a method described in the appendix, of a set of coefficients characterizing the given linkage. The friction terms introduced in the calculations depend on the load and speed conditions and must be evaluated empirically. These terms must be taken into account if the calculations are to yield reliable results. Experimental results have shown reasonably good agreement with calculations. If an automatic computer is available, the amount of calculation involved is not excessive. *M.I.R.A. Abstract No.* 6276.

PLASTICS REFERENCES

IN an article entitled "Selected Plastics References for the Mechanical Engineer 1951-1952," by A. K. Schmeider and H. G. Dikeman, in the June 1953 issue of *Mechanical Engineering*, a list of references is given, some of which may be of use to automobile engineers interested in laminated plastics for motor bodies. Among them are those given below, which are mainly concerned with aircraft structures; however, many of the problems apply to the automobile industry.

"Low Pressure Laminates for Aircraft," *British Plastics*, vol. XXIV, October, 1951.

"Moulded Wings for Delta Aircraft," *British Plastics*, vol. XXIV, October, 1951.

"Designing with Glass-Reinforced Plastics," *Product Engineering*, vol. XXII, November, 1951.

"Aircraft Plastics Structures," by H. W. Hall and J. E. Gordon, *Aircraft Production*, vol. XIII, July, 1951.

"Plastic Car Body in Production," *Modern Plastics*, vol. XXIX, April, 1952.

"Compound Curved Honeycomb Core Material for Sandwich Panel Construction," by A. H. Herbst, *Society of Plastics Engineers Journal*, vol. VIII, January, 1952. (2050)

GAS TURBINE ARRANGEMENTS

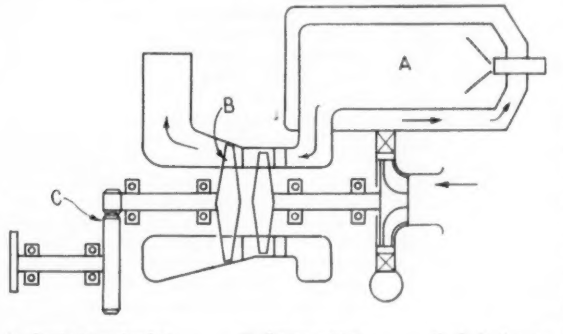
A Review of Systems for Automobiles

F. R. Bell

A PREVIOUS article in the February 1953 issue of this journal entitled "The Problems of the Turbine Car" discussed the design of the car itself, the transmission and the modifications necessary to a small gas turbine to make it suitable for installation in a car. It is proposed here to consider the various types and arrangements of gas turbine that may be used in a car and give the reasons for the final choice. It should be mentioned that these arguments only apply to the gas turbine for use in an automobile.

In general the choice has been for simplicity; thus where either of two designs could be used, the simpler has been chosen unless there is a fairly strong case for the more complex type. This results in more rapid and cheaper development although there is the possibility that at some time in the future it may turn out that one of the other designs will be better.

Firstly, it will be shown that the free power turbine is the obvious choice; secondly, the chosen engine, which will be called the standard design, will be described; and thirdly the alternative engines will be described and compared with the chosen engine to show why they are not so suitable.



A—Combustion chamber. B—Power turbine. C—Reduction gear

Fig. 1. Free power turbine ; power turbine left, compressor (jet) unit right

The free power turbine

Although this is obviously the only sensible arrangement it is not, perhaps, well understood by everyone and the arguments for and against should be given.

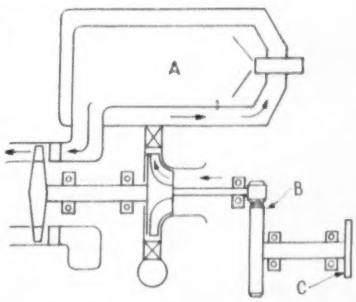
In the free power turbine system, the engine is comprised essentially of two parts: the power turbine and the gas generator. These are not connected mechanically. As shown diagrammatically in Fig. 1, the gas generator consists of a compressor, turbine and combustion chamber very similar to an aircraft jet engine. The exhaust or jet from this flows through the power turbine, the shaft of which is not connected to the compressor turbine. The car is driven from the power turbine through gearing. The alternative would be to take the drive direct from the compressor turbine shaft as shown diagrammatically in Fig. 2.

With the free power turbine, the compressor unit can be started with the car stopped, but with the compressor unit running at idling speed the gas flow through the power turbine is insufficient to apply any noticeable torque. It is in fact insufficient to move the car even without the handbrake on. When the compressor unit is accelerated the gas pressure temperature and flow are increased to give an increased torque and so start the car. Actually when the compressor unit is run up to full speed the torque is $2\frac{1}{2}$ to 3 times the value at full car speed. As this is somewhere between the first and second gear torque of the piston-engine car it can be seen that either no gear box at all or only a two-speed for emergency will be needed.

In the alternative system, the torque characteristics are actually much worse than for a piston engine and therefore a gearbox with probably five or more speeds will be necessary. Curves in Fig. 3 show these relative characteristics. As the ability to drive without any gear changing is the main advantage of the gas turbine it can be clearly seen that this engine arrangement would be utterly useless. For this reason, in all except one of the designs considered here a free power turbine arrangement will be used.

The standard design

The arrangement chosen is shown diagrammatically in Fig. 4. This is probably the simplest possible arrangement and consists of a simple single stage single sided centrifugal compressor with diffuser and a single stage axial flow turbine driving the compressor. The air passes first through the centrifugal compressor and the air side of the heat exchanger, then to the combustion chamber and compressor turbine. The exhaust from this passes through the single stage power turbine and gas side of the heat exchanger and out to atmosphere.



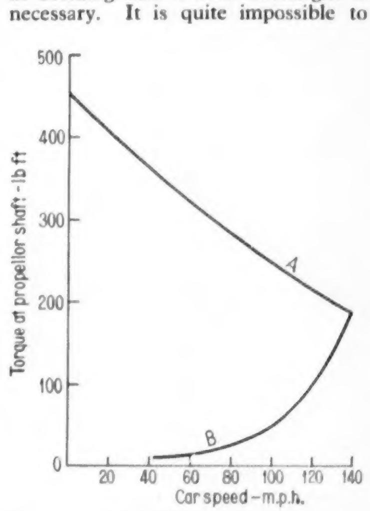
A—Combustion chamber. B—Reduction gear. C—Power output.

Fig. 2. Direct drive engine

Auxiliary drives are taken from the inlet end of the compressor via a reduction gear and at the opposite end of the engine is the power turbine reduction gear. Heat exchangers of the sheet metal matrix type are arranged on both sides of the engine with the combustion chamber on top. Each turbine has a stationary row of blades or nozzles and a rotor with the blades manufactured integrally.

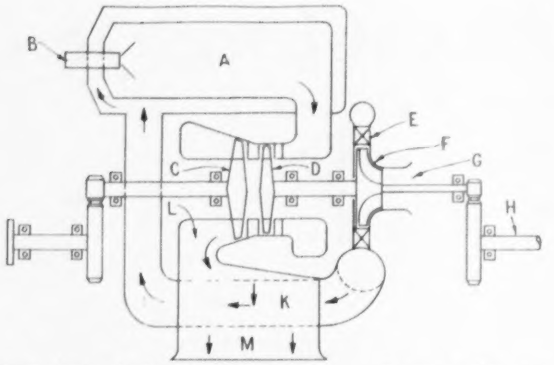
A single stage reduction gear is used and the power turbine is arranged to rotate in the opposite direction to the compressor turbine. Maximum compressor speed is such that the compressor tip speed is about 1,500 ft/sec.

At this point it might be as well to insert a word about the difficulty of deciding just how big a heat exchanger should be. There is no difficulty at all in deciding that a heat exchanger is necessary. It is quite impossible to



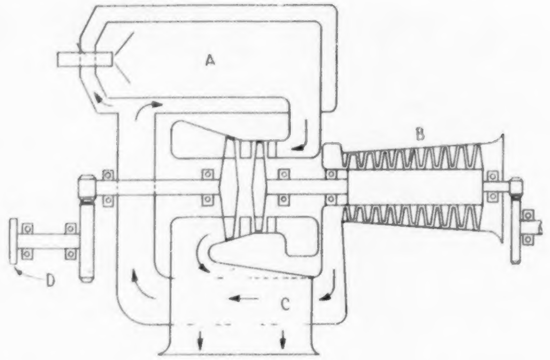
A—Torque at 4,000 r.p.m. with free power turbine. B—Torque without free power turbine.

Fig. 3. Comparative torque curves



A—Combustion chamber. B—Fuel nozzle. C—Power turbine rotor. D—Compressor turbine rotor. E—Diffuser. F—Compressor impeller. G—Air intake. H—Auxiliary drives. K—Heat exchanger. L—Power turbine diffuser.

Fig. 4. Standard engine

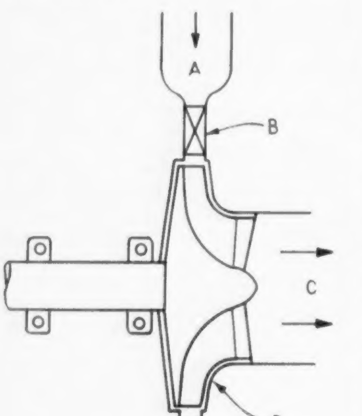


A—Combustion chamber. B—Axial compressor. C—Heat exchanger. D—Power output.

Fig. 5. Axial compressor engine

get a sufficiently good efficiency, especially at the low power end, to make a commercial success of the gas turbine car without one.

The bigger the heat exchanger the more efficient the engine becomes but as the gains get progressively less with increasing size there will obviously be an economical limit. This effect is made still worse by the fact that as the heat exchanger gets larger it makes the whole car heavier and so spoils the fuel consumption. In the limit, with an infinite size of heat exchanger the fuel consumption will be no miles per gallon since the car will not be able to move itself even though the engine efficiency is at its maximum value which is, however, not more than about three times the value without heat exchanger. Because this decision will primarily rest with the buyer, that is it will depend on how much extra he is prepared to spend on the first cost of a car in order to obtain a higher fuel consumption, it cannot be accurately decided until there are a number of cars on the road.



A—Gas inlet. B—Nozzles. C—Exhaust. D—Rotor.

Fig. 6. Radial flow turbine

Size will also be affected by the cost of production of the heat exchanger. Obviously, if the heat exchanger is very cheap to manufacture, a small additional cost will give a very large heat exchanger, but if it is very expensive then only quite a small size will be used.

Of course it may well be that, with the high cost of fuel, people may look well ahead and say that in the life of the car the saving in fuel due to a larger heat exchanger is justified. In this case the size will be limited primarily by the space available, but the life of the heat exchangers must also be taken into account because the cost of replacements will obviously be greater with a larger heat exchanger. The life can only be found under actual running conditions when the cars are in production.

Axial compressor engines

The first alternative to be considered is whether to use an axial compressor. An engine using an axial flow compressor is shown diagrammatically in Fig. 5. Probably the most important reason for not using the axial compressor is its sensitivity to dirt deposits on the blading. It has been found that aircraft axial compressor engines when run on the ground build up a deposit of dirt on the leading edges and backs of the blades causing a serious reduction in efficiency. Either filters must be used or the dirt must be removed by frequent cleaning by spraying paraffin or some other suitable liquid into the intake. This trouble does not occur while flying as there is practically no dust at high altitudes.

Frequent cleaning in a car would be a serious disadvantage and owing to the very large mass flow of the gas turbine the use of filters is rather impracticable. The other bad disadvantage of the axial compressor is its high cost. Even a small compressor for use in a car would probably have 500 to 1,000 blades and it is extremely unlikely that these can be made for anything like the cost of the simple centrifugal compressor.

Four reasons make the axial compressor the most suitable for aircraft use. Firstly, it has a very much smaller frontal area for a given mass flow of air. Secondly, it runs faster for a given mass flow, thus reducing the required turbine diameter, since the turbine must run at the same tip speed in both cases. Thirdly, it can conveniently be made to a higher pressure ratio than a single stage centrifugal, and fourthly it has a higher efficiency than the centrifugal. Except for the last none of these reasons is much help to the car turbine and in the case of the last, it is unlikely that the very small axial compressor, owing to its very small blades, would have even as good an efficiency as the small centrifugal. Taking into account the fact that the small axial compressor is completely undeveloped, it is obvious that it is much inferior to the centrifugal, at least for the present.

The radial flow turbine

A radial flow turbine, which is in principle simply a centrifugal compressor with inward gas flow, is shown in Fig. 6. As this is a very controversial

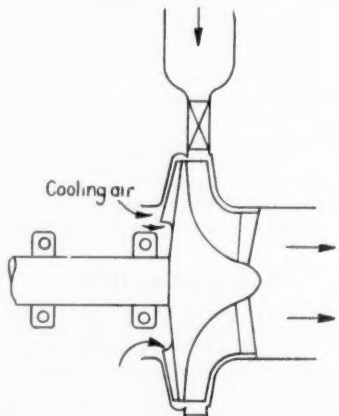
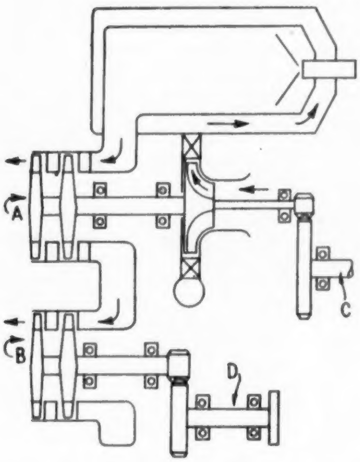


Fig. 7. Radial flow turbine with cooling arrangements

subject, it is not proposed to go into it too deeply or dogmatically. It may turn out finally that the radial flow turbine is actually the best but this is rather unlikely. There will only be space in this article to discuss briefly some of the more important points.

It is often claimed that the radial flow turbine, particularly in small sizes, is more efficient than the axial flow. At present there is absolutely no evidence of this, in fact there is some small amount of evidence to the contrary, the most important perhaps being that the French-made Turbo-méca small gas turbine has, under government test, already apparently given considerably higher turbine efficiencies with axial flow turbines than has any radial flow turbine yet tested. The claim for higher efficiency has apparently come about partly through the misreading of certain test results from a radial flow turbine test rig and partly from comparing the best of the radial flow figures with some axial turbine results not of the best.

The second claim of the defenders



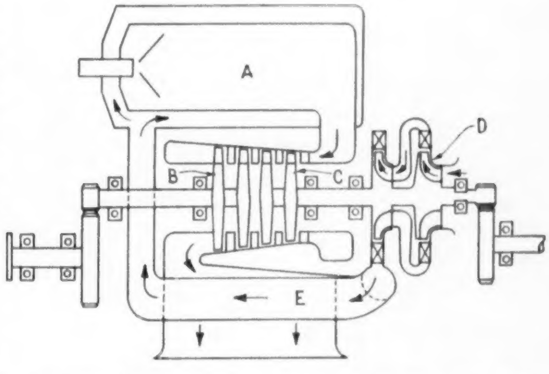
A—Compressor turbine. B—Power turbine.
C—Auxiliary drive. D—Power output.

Fig. 9. Divided flow engine, shown without heat exchanger

of the radial flow turbine is that it should be cheaper to manufacture. Cost comparisons are rather difficult because at present we do not know which method of manufacture is likely to be the best. In making the comparison it is essential to compare turbines doing equal jobs. An engine can be made either with a fairly large air flow and low pressure ratio which allows low turbine speed and hence low turbine stresses, or with a smaller mass flow and higher turbine stresses owing to the higher turbine speeds necessary to produce the same power.

If the first type of engine is used the efficiency will be lower and the engine weight greater but cast turbine rotors can be used for either axial or radial flow turbines. Nozzles in all cases will probably be cast as they are practically unstressed compared with the rotors.

There should be practically no difference between the cost of casting the rotors in either case; a slightly simpler casting for the radial flow being offset by about fifty per cent more material being used in the radial flow rotor, the materials being the same in both cases. If the higher speed, higher efficiency engine is used, and for a car where the efficiency is barely high enough in any case it is almost certain to be, then the rotors in both cases will probably have to be forged.



A—Combustion chamber. B—Two stage power turbine.
C—Two stage compressor turbine. D—Two stage compressor. E—Heat exchanger.

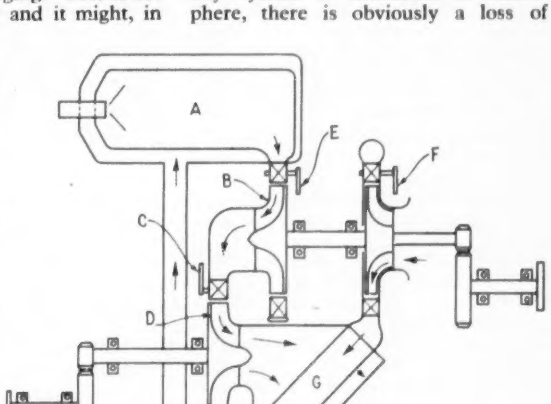
Fig. 8. Two stage engine

Actually, because the acceleration time of the compressor turbine unit is important in a car and because this is proportional to the moment of inertia, it is essential to keep the turbine weight to a minimum. This means that even for the first type of engine it is advisable to use forged rotors. Also, as the weight of the radial flow turbine seems to come out higher in the actual designs considered so far, it would seem preferable to use axial flow.

If forged rotor material is used, it would be far too expensive to machine either axial or radial rotor from the solid so that we only need to compare the rotor forged nearly to size. Experiments with forging have shown that the axial rotor complete with blades can be forged in one operation fairly easily. This is because the blades are small in relation to the size of the forging and therefore the metal has very little distance to flow, a very important point in forging. This is not so in the radial flow and it might, in fact, be more difficult to forge than the axial. Both types, owing to the necessity for a good finish for air flow, will probably require a final machining. The axial flow type has only about one-half of the blade surface to be machined but the cutters cannot be so rigid. Probably the machining times, judging from tests made with special milling machines on the blades of axial turbines, will be about the same in each case.

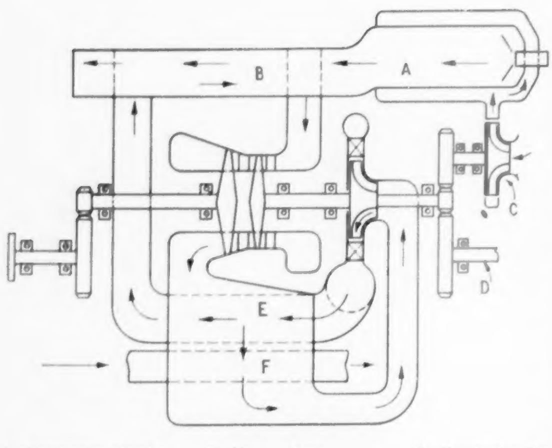
Cooling is the next question that arises. The ease with which cooling air can be blown up the back of the impellor has been stated to give a big advantage to the radial flow from the point of view of cooling. This is not so for several reasons, in fact the reverse may very well be true.

Firstly, it must be realized that any form of cooling entails a loss of efficiency, that is, using a fixed gas temperature, the more cooling that is used to make the rotor cool enough so that it will not fail, the lower the efficiency. This applies whether the air is bled from the compressor or pumped by a separate fan. If blades are on the back of the turbine rotor like another compressor, as shown in Fig. 7, then heat is fed into the air during compression and out of the turbine during expansion, lowering both the turbine and compressor efficiency. This means that more power is required on the compressor blades than is obtained from the cooling air flowing through the turbine, with a consequent loss of power. If, after cooling the back of the disc, the air on any system is exhausted to atmosphere, there is obviously a loss of



A—Combustion chamber. B—Radial flow compressor turbine.
C—Power turbine nozzle control. D—Radial flow power turbine.
E—Compressor turbine nozzle control. F—Diffuser vane control.
G—Heat exchanger.

Fig. 10. Radial flow variable nozzle engine



A—Combustion chamber. B—Heat exchanger. C—Combustion fan. D—Auxiliary drive. E—Heat exchanger. F—Cooler.

Fig. 11. Closed circuit engine

heat which means a loss of efficiency.

The only system without significant loss is to take air from the compressor delivery and, after cooling the back of the rotor, feed it back into the turbine inlet. The difficulty here is that it is almost impossible to make a seal at the turbine rotor tip because, owing to the thermal expansions and bearing slack, the clearances have to be so great that even with a labyrinth the leakage is excessive. Tests have shown this very definitely to be the case.

As all these arguments apply equally to both the axial and radial flow turbines, and as in the radial flow turbine the total blade and disc surface exposed to hot gas is something like twice as great, it may need twice the cooling air to bring the metal temperature to the same value.

Primarily, these arguments have been related to the compressor turbine but generally they apply equally well to the power turbine. For the power turbine there is, however, a further argument against the radial flow. Owing to the variation of the centrifugal pressure at the rotor tip with speed, the radial flow power turbine will cause a back pressure on the compressor turbine varying with car speed. At a given compressor speed, this causes the temperatures to rise at high car speeds and fall at low car speeds. At low car speeds the compressor tends to run on to a part of its curve where the efficiency is lower.

When the car is stopped and the compressor is taken to maximum speed the temperature is low, therefore the power is down and the acceleration of the car is poor. If the engine is matched to be hot at these conditions it will be too hot at full car speed and will therefore have to have its compressor speed limited thereby lowering its efficiency.

Variable nozzles for the power turbine may, of course, overcome this difficulty, but they are an added complication and will be very difficult to make work satisfactorily. In any case, if the radial flow turbine requires variable nozzles and the axial flow does not then there is not the slightest doubt that the axial flow turbine is the better.

The multi-stage engine

This is shown diagrammatically in Fig. 8. There are actually a number of very good arguments in favour of the multi-stage compressor and turbines and the most important argument against is simply that it is very much more difficult to make and to make work. If a multi-stage compressor is used it will have two stages, and there are two ways of using it. Either it is run at a speed to give the same overall compression ratio or it is run at a speed to give the same rotor tip speed. There is very little point in running at the same overall compression ratio, as the speed will be lower, the turbine therefore larger, and its shorter blades for the same annulus area may lower its efficiency as much as the compressor efficiency is raised by being two stage. Of course, the turbine can also be two stage, but there is no evidence that for compression ratios of 4:1 or less that a two stage turbine is any more efficient than a single stage. In effect, there is no point in running at the same compression ratio with two stage as the cost is nearly double for a negligible gain in efficiency. If the compressor is run at a higher compression ratio, say 5:1 or 6:1, then a two stage turbine will be necessary to drive it. As the pressure ratio available at the power turbine will also be higher, a two stage power turbine may also be necessary.

Perhaps it should be pointed out here, that the number of stages used in a turbine depends primarily on the pressure drop available across the turbine. A single stage turbine of the type used in gas turbines can fairly efficiently deal with a pressure ratio of up to about 3:1, but by using two stages for the pressure ratio the blade speed and hence the blade stresses will be lower. Although this will allow cheaper material to be used, the overall cost will probably be higher and certainly the weight will be greater, and there is no evidence to suggest much if any increase in efficiency. For a power turbine, owing to the necessity for a high torque ratio, there is some evidence that it is not advisable to use pressure ratios greater than 2:1 in a single stage.

Considering the two stage compressor and two stage turbine type of engine running up to say 6:1 compression ratio, the fuel consumption will undoubtedly be better, particularly at low speeds, if the same sized heat exchanger is used as with the single stage. The engine will, however, cost maybe twice as much to make as it has twice as many of the most expensive components and is considerably more difficult to deal with for alignment and clearances.

If the same amount of money were spent on a larger heat exchanger for a single stage engine it would probably give as good an efficiency and be much simpler. From the servicing viewpoint the two stage engine is obviously much more difficult. Apart from

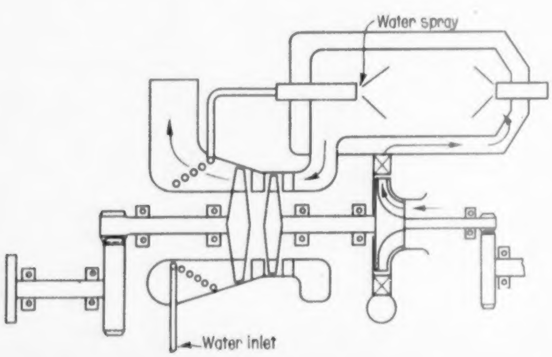
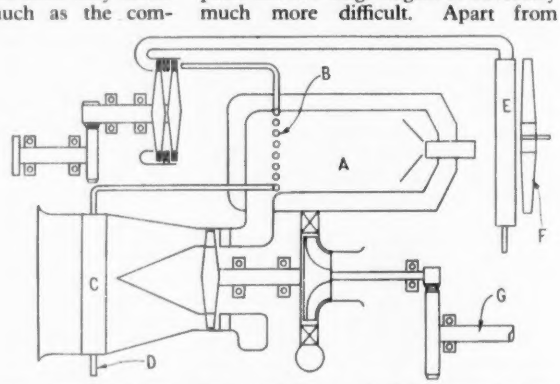
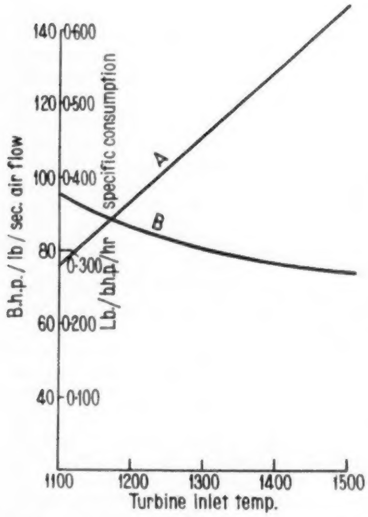


Fig. 12. Mixed cycle engine, water injection



A—Combustion chamber. B—Superheater. C—Heat exchanger (boiler). D—Water feed. E—Condenser. F—Cooling fan. G—Auxiliary drive.

Fig. 13. Mixed cycle engine, steam turbine



A—b.h.p/lb/sec. B—Specific consumption.
Fig. 14. Effect of turbine inlet temperature on efficiency

improved fuel consumption there is no reason at all for making the two stage engine.

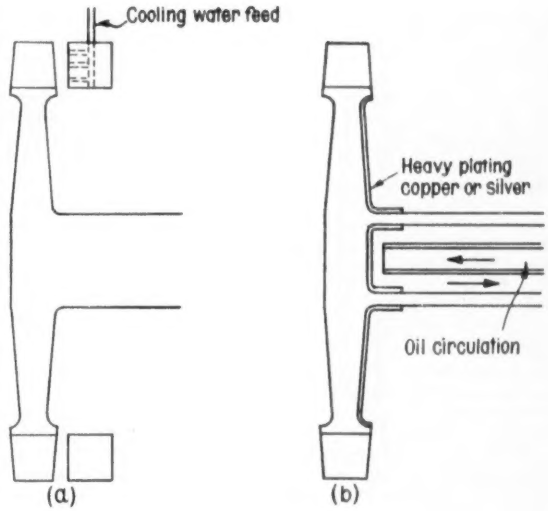
Divided flow engine

Another system, which is shown diagrammatically in Fig. 9 and which must be mentioned is the divided or parallel flow. In this system the compressor turbine exhausts through the heat exchanger to atmosphere and the air for the power turbine is bled off either after the combustion chamber and through the power turbine and heat exchanger to atmosphere, or before the combustion chamber so that a separate combustion chamber may be used. This system has no merits at all for car use; there is no gain in efficiency and each turbine requires to be two stage as each has the total pressure drop to cope with. Also, as each has less than the total mass flow it will be a smaller turbine and therefore, probably of a lower efficiency. It can easily be seen that this system offers no advantages at all.

Variable nozzles and diffuser

Two main ways are available in which we may vary the nozzles or diffuser, that is either by turning the blades to alter the angle or by closing some passages. This last method on turbines is called partial admission, and, as it results in quite a considerable fall in turbine efficiency, that is, about 10 per cent, it is quite useless for the gas turbine. If variable nozzles are used they are more suitable for use with radial flow turbines, therefore the radial flow engine would probably be used. A diagrammatic arrangement is shown in Fig. 10.

The main argument in favour of variable nozzles is that if the diffuser and both turbine nozzles are varied the engine can have a variable mass flow. This, it is said, will allow the mass flow to be reduced to obtain low powers, thus keeping the compression ratio and temperature higher and giving a better efficiency. It is rather doubtful if this is true, because owing to the severe thermal shock on compressor turbine nozzles, quite large clearances will be necessary to prevent jamming. This and the fact that angles will not always be quite right for the best efficiency, may lower the efficiency overall more than has been gained by the reduced flow. If there is no gain or only a small gain in efficiency then this type will certainly not be justified.



(a)—Water cooled (spray). (b)—Oil cooled by conductivity.
Fig. 15. Cooling systems

Even if there is a significant gain in efficiency, the mechanical difficulties are very great. Running engines at high temperatures has shown that compressor turbine nozzles suffer severely from cracking and distortion even in the best materials. Slack fits will be necessary to prevent sticking, and heat-resisting steels are notoriously bad at picking up, particularly as no lubrication can be used and the high frequency vibration from air flow will cause severe fretting. Still, it may be possible to make variable nozzles and diffuser work satisfactorily but they will be rather expensive for possibly a not very large gain in efficiency. Again it may pay to spend the money on a much larger heat exchanger.

Closed cycles

In the closed cycle system the heat is fed into the working fluid through a heat exchanger instead of a combustion chamber and the turbine exhaust is then sent through a cooler to reject heat and back into the intake. The working fluid is therefore circulating continuously and can be sealed in. This arrangement is shown diagrammatically in Fig. 11.

Because the fluid is sealed in, various different gases can be used and there are all sorts of advantages and disadvantages for each. Several different types occur. Inert gases like neon, which will not attack the metals used in the turbine even at high temperature, make the manufacture of the engine easier because cheaper materials may be used. The monatomic gases allow lower compression ratios to be used. The heavy gases allow less volume flow to be used and therefore the engine can be smaller. Variable specific heats (sometimes due to dissociation) may have advantages as they make the compression and expansion cycles different. Some gases are too expensive. Actually, all the requirements are so conflicting that the conclusion must be that air is the best all-round medium.

Probably the main advantage of this system is the ability to vary the pressure in the unit. This is in effect a way of varying the mass flow and hence the power. A car would probably be controlled by varying the pressure instead of the speed of the unit, thus giving practically constant efficiency.

Unfortunately there is one very serious disadvantage not obvious at first sight. A gas turbine is very sensitive to intake temperature and it is not possible without an extremely efficient cooler to get this temperature low enough for even reasonable efficiency except perhaps for power stations and marine work where large quantities of cooling water are available. There is the added disadvantage that now there are three heat exchangers (one is difficult enough) and some sort of auxiliary fan for combustion air.

Mixed cycles

Shown diagrammatically in Figs. 12 and 13 are two varieties of the mixed cycle engine. Fig. 12 shows water injected into the combustion chamber. This scheme is of no use unless there is ample supply of fresh water. It is likely to have many important uses but certainly not in road vehicles.

The second scheme feeds exhaust heat into a steam turbine. It is really a steam turbine system and could give very good efficiency but would probably be excessively complicated and expensive. Both these schemes have modifications and variations far too numerous to give here but none are likely to have any useful place in the automotive world.

Using heat exchangers in an engine giving shaft horse power, very large gains in efficiency occur with higher

temperatures as shown in the curves in Fig. 14. Because of this, the standard simple single stage engine with good heat exchangers, if it can be made to withstand high enough temperatures, can give very high efficiency.

With these high temperatures, difficulties occur mainly with the turbine blades, but also with the combustion equipment and heat exchangers. Heat exchangers and combustion equipment can almost certainly be made suitable with careful detail development, but it may be necessary to cool the turbine blades. It is rather unlikely that the air-cooling methods using hollow or drilled blades will ever be of much use for small turbines as they will be much too expensive for small blades, but they may possibly be used if someone can solve the problems of manufacture.

Quite a number of cooling methods have been suggested and two may be taken as examples. If carrying and using a small amount of water is not too much of a disadvantage, say equal to 10 per cent of the fuel, then water sprayed on to the blades is simple and effective.

In the very small turbine with integral blades, the conductivity is good enough so that heat is fed down to the shaft and out with the oil giving quite useful cooling. This may be considerably improved by improving the conductivity of the disc by plating, or by drilling holes for oil or sodium and by having lavish oil flow and oil coolers. Some of these schemes are shown diagrammatically in Fig. 15.

Other types

Schemes to improve the efficiency of the gas turbine for automotive use have been exercising the minds of inventors ever since gas turbines have been thought of and particularly since the Rover Co. showed their lead over the world in this field by demonstrating the turbine car. Hundreds of suggestions were put up to the Rover Co. but not one original suggestion had any real merit. Most showed a complete lack of understanding of the difficulties involved and some were, of course, engine arrangements already well known and discarded.

All work so far has indicated that the problem is not one of inventing some new secret scheme but of laboriously improving the component efficiencies of the simple standard turbine design. It is impossible in an article of this nature to cover all engine arrangements that can be thought up, but enough types should have been discussed to give a general idea of the problems involved.

It may be concluded from the foregoing discussions, that while it is impossible to foresee the type of gas turbine that will finally be evolved for use in the motor car, it is best at present to keep to the simpler types. These should give results almost as good as the more complicated types and should be much cheaper and quicker to develop and manufacture.

CORRESPONDENCE

AUTOMOBILE DYNAMIC LOADS

SIR,—In his article "Automobile Dynamic Loads" in the February issue, Mr. Garrett has endeavoured to put forward a logical method of determining the loads to be used in stressing chassis parts, using a system based upon standard aircraft practice. This aim is entirely to be commended, but the present writer feels there are some points that could well have been clarified, particularly as the majority of readers will probably be unfamiliar with aircraft practice. Further, the application of this method to automobiles appears to have certain limitations.

The conditions which cause the highest dynamic loading in aircraft, whether due to a manœuvre or an upward or downward gust, persist long enough to cause bodily accelerations of the aircraft (and its passengers) as a whole, so that the assumption of a certain value of "g" applicable to the whole machine is justifiable. On the other hand, with an automobile the bump loading is of very short duration and is not transmitted directly to the chassis owing to the springing. The forces which *are* transmitted produce two motions; a low-frequency bodily motion (pitching or bouncing), upon which is superimposed high-frequency elastic vibrations of the chassis. The latter, though not felt by the passengers, are those which give rise to the high stresses. The pitch and bounce motion only gives rise to quite small stresses—after all, the passengers do not have to be strapped in when going over a rough road, as they would if the bodily acceleration were anything like ± 3g. It is nearer to the value corresponding to the threshold of discomfort, which is about $\pm \frac{1}{4}$g at normal car frequencies of springing.

The vibrational acceleration (of the order of ± 3g) thus depends very much upon the elastic properties of the springing and the chassis, and cannot be directly related to a given shape of bump, as can be done in relating aircraft accelerations to gust intensities, since elasticity of the aircraft structure has comparatively small effect.

The following conclusions can therefore be drawn:—

(1) Whatever the nature of the dynamic loads, they are superimposed upon the static load due to dead weight, which is always present. To quote Mr. Garrett's own example, a petrol tank with an upward acceleration of 48 ft/sec/sec ($1\frac{1}{2}$g) will have a load on its mountings of:—

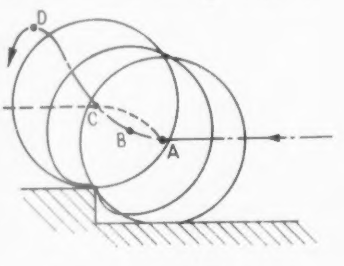
$1\frac{1}{2}$W dynamic + W static = $2\frac{1}{2}$W.

If the acceleration had been downwards the load would be:—

$-1\frac{1}{2}$W dynamic + W static = $-\frac{1}{2}$W (downwards).

The basic assumption of dynamic loads of ± 3W will therefore give total loads of +4W to −2W. The factor of safety of 1·5, giving design loads of 4·5W, therefore gives a margin far less than has been assumed. Neglect of the static loads affects much of the remainder of the article.

(2) In the case of an automobile, the accelerations due to an impact, being vibrational, will vary along the chassis. However, unless this variation can be shown to be considerable, the assumption of a uniform value along the chassis is justified on the grounds of simplicity.



A Initial contact with bump
B Maximum ground reaction
C Wheel leaves ground
D Maximum reaction on to chassis
— —— —— — Path of wheel centre
— — — — — — Path of wheel centre if there is no elasticity or rebound of the system

(3) The simple geometrical relationship between vertical and rearward loads due to a bump, as given in Fig. 2 of the article, does not apply for two reasons:—

(a) The vertical forces should include the dead weight, whereas the rearward ones do not.

(b) The minimum vertical and rearward forces occur at different times and there is no simple relationship, as shown by the accompanying figure. The maximum wheel load, and hence the rearward force on the chassis, occurs at point B, whereas the maximum reaction vertically on the chassis is at point D. Expressed mathematically, if P is the upward ground force at any instant, W the unsprung weight and v its upward velocity as it leaves the ground at point C,

$$\frac{W}{g}.v = \int P.dt$$

equating the impulse to the upward momentum. This momentum is destroyed by a combination of spring, snubber and damper loading, so that if R is the upward reaction on the chassis, we get

$$\frac{W}{g}.v = \int R.dt$$

Hence R can only be equated to P if the duration of the bump is equal to that of the snubbing action, which is not necessarily true. A separate basis for the rearward loads is required.

A further point about the proposal for a standard bump, is that a more severe case than a kerb would be a depression having a curved end of the same radius as the tyre. The load on the tyre is then distributed over a large arc, and tyre flexibility has a minimum cushioning effect.

B. G. de BRAY,
M.I.Mech.E., A.F.R.Ae.S.,
Senior Lecturer,
Mechanical Engineering Dept.,
School of Engineering,
Auckland University College,
Auckland, N.Z.

Mr. Garrett replies: —

With regard to the question of the duration of load, mentioned in Mr. de Bray's letter, the vertical accelerations referred to in the article were obtained from tests with an accelerometer mounted in the vehicle. It can hardly be disputed therefore that loads of this order may be imposed on the structure. Admittedly, accelerations of the order of 3g are, in fact, infrequently encountered, but an owner would feel justifiably aggrieved if structural failure occurred on the few occasions when the vehicle was subject to unusually severe conditions.

Differences of opinion might arise as to the severity of load that should be catered for, and future investigations may prove that 3g is too high. However, by checking mathematically the stresses that would be obtained when applying this load to existing designs, it can be shown that current practice is to make the main structural members of such a size that they are capable of withstanding loads of this order. It is therefore felt that it would be unwise to recommend designing for lighter loads until more experimental work has been carried out on this subject and positive proof has been obtained to justify such a step.

I agree with the correspondent that the high frequency vibrations of chassis frames are of considerable importance. That they are more important than is generally realized was mentioned in my contribution to the discussion on the paper entitled "Life Assessment Tests for Commercial Vehicles," by John Alden, read before the Automobile Division of the Institution of Mechanical Engineers in London on the 13th January 1953. However, the stresses induced both by these vibrations and by the frequently recurring loads of about ½g referred to by the correspondent can be serious only from the point of view of fatigue failures. Practical experience and theoretical considerations suggest that vehicles designed for the maximum loads quoted in the article will have adequate fatigue strength so far as the light loads encountered in normal service are concerned.

With regard to item (1) of the correspondent's conclusions, the load of 1¾W quoted in the article referred, of course, to the dynamic loads only, as was stated in the first line of the paragraph in which it was quoted. However, some confusion may have arisen through the omission of the static load and I am grateful to Mr. de Bray for clarifying this point.

The vertical load of $\pm 3W$ was adopted as a result of a study of research reports describing practical experimental work. One such report was described in the article as "the experimental work on commercial vehicles." Unfortunately, this report has not been released for general publication so I cannot be more specific. It gives details of stresses induced in motor vehicle structures under various conditions of dynamic loading, and from it I drew the conclusion that, until more is known about the subject, it would be safest to assume that the plus and minus values of the vertical loading are equal. The value of 3g was decided upon not only because it was in general agreement with the findings of a number of investigators but also because it appeared to be consistent with current design practice. However, I agree with Mr. de Bray that this load may not be accurately representative of the true conditions, but as was stated in the article, it is considered advisable to use it until more information is available.

Item (3) in the letter refers to the simple geometric relationship between vertical and rearward loads due to a bump. In answer to point (a), I maintain that the rearward load is the horizontal component of a resultant force, the vertical component of which includes both dead weight and acceleration forces; therefore it also includes the corresponding proportions of dead weight and acceleration forces.

Point (b) raised by the correspondent, and his illustration, are only applicable if the full bump position is not attained. However, under the severe conditions that give rise to vertical accelerations of 3g, the suspension will certainly be up against the bump stop and the maximum vertical and rearward loads under these conditions will be applied simultaneously. Moreover, neglecting rotational inertia effects, the wheel may be assumed to pivot freely about its axis, so that the rearward and vertical loads must be interdependent, because the resultant of the two components passes through a common point on the wheel axis.

Certainly a depression having a curved end of the same radius as that of the tyre would tend to give, at the first instant of impact, a more severe load than a kerb. However, the severity of this first shock will be reduced by the deflection of the suspension as it rides up over the edge of the depression, and the maximum load is likely to occur when the bump stop is struck.

FUEL INJECTION NOZZLES

Sir,—I read with much interest the article in your May issue entitled "Fuel Injection Nozzles." I think your readers may be interested to know that I carried out a similar investigation about 1938 and that the result brought to light a point which is not mentioned by Mr. Mansfield in your article.

A customer sent us nozzles showing erosion just similar to that illustrated in the figures in your article. We were fortunate in having an engine available of the same model as that in which the trouble was occurring. Our first step was to run tests with outlet jacket water temperatures as low as 80 deg F, but these produced no effect.

The next step was to form grooves across the face of the copper joint washer so that blowing took place which could be felt distinctly at the upper end of the nozzle holder, and which was sufficient to make starting difficult. Again there was no erosion.

Finally the diameter of the nozzle was reduced over the shank where it fits into the cylinder head so that there was a diametral clearance of 1 mm. Under these conditions, in conjunction with a cooling water outlet temperature of 80 deg F and running at no load, erosion was produced exactly similar to that on the nozzles which had been submitted to us with the original complaint.

It would be interesting to have Mr. Mansfield's comments and to know what was the clearance between the nozzle and bore in the cylinder head in the case of his experiments.

W. A. GREEN.

Mr. Mansfield replies: —

Mr. Green's account of his experience with nozzle corrosion is most interesting, and is further evidence of the widespread occurrence of this trouble.

In our own work we gave consideration to the effect of the diametral clearance between the nozzle and the bore in the cylinder head, but did not find it necessary to alter this from its original design value in order deliberately to produce corrosion. Later we investigated the effect of increasing this clearance because we thought this might be a suitable method of increasing the nozzle temperature, by virtue of the greater movement of hot gases along the sides of the nozzle, and hence of reducing corrosion. A preliminary test had shown that increasing the mean diametral clearance from 0·30 mm to 0·45 mm by reducing the diameters of the nozzle stem (which tapered initially from 9·2 mm at the end adjacent to the cap-nut to 9·0 mm at the discharge end) caused a reduction of 7 deg C in the temperature of the copper washer surrounding the cap-nut end of the nozzle stem. This was attributed to the reduction in the surface area exposed to the hot gases and the reduction in the section of the metal conveying heat to the cooler end. The effect may account to some extent for Mr. Green's finding.

In view of this result, we made a series of tests on another engine using a new standard nozzle for each test, and progressively increasing the clearance by enlarging the bore in the cylinder head. The diametral clearances used were 0·425 mm (standard), 1·425 mm, 1·925 mm and 2·925 mm, and the corresponding losses in weight of the nozzles in 30-hour runs were 70 mg, 23 mg, 40 mg and 25 mg. A check test with the 1·925 mm clearance gave a loss of weight of 73 mg. This wide scatter was thought to be associated with the extent of carbon deposition which varied considerably.

In some later tests, separate water-cooled test pieces were used instead of nozzles. After tests with the Class A fuel previously used, the engine was run on a Class B fuel having a much higher sulphur content which was expected to result in an increased rate of corrosion. No measurable loss of weight occurred. The clearance space adjacent to the combustion chamber was filled with very hard carbon which made removal of the test piece difficult. It was concluded that this carbon had protected the cool surfaces of the test-piece from the combustion gases.

The rapidity with which the clearance space becomes filled with carbon may depend on the size of the clearance used, as well as on the fuel. It may be that in Mr. Green's test the rate of carbon formation and the nature of the carbon formed were such as to seal the smaller clearance first used. If the nozzle were made a sufficiently tight fit in its bore, there would be little likelihood of attack, but this, of course, would probably cause distortion of the nozzle.

In all the tests described above, the end of the nozzle remote from the combustion chamber was excessively cooled for the purpose of the experiments. We considered that the supplementary findings supported the conclusion given in our report, that the best practical solution of the problem is to avoid excessive removal of heat from the nozzle, either by avoiding very low inlet water temperatures or by a suitable change in design of the cooling passages.

CRANKSHAFTS

Laystall Engineering Co. Ltd. have recently doubled the output potential of their crankshaft factory. The shafts that can be produced range from small ones, 6 in long, to large ones 7 ft long × 9 in stroke for marine and stationary diesel engines.

DIESEL ENGINE PRODUCTION

A Survey of the Equipment and Methods Employed by Frank Perkins Limited

THERE is a generally held opinion that in the production of internal combustion engines American output is in all cases far higher than British output. It is, therefore, not without point to emphasize at the outset of these notes that Frank Perkins Ltd., Peterborough, produce more engines in the range with which they deal than any organization in the U.S.A. In fact, we believe that over this range the Perkins output is greater than the output from all the American factories. In its twenty-one years' existence this organization has shown a remarkable growth. The years before the war saw a slow but continual increase in output; since the war, the increase has been spectacular.

In 1946 the output was at the rate of 3,500 engines per annum, but the demand was so greatly in excess of production potential that a new factory of 120,000 ft^2 area, plus auxiliary buildings, was planned. This development was completed in 1947, but it soon proved insufficient to cope with the rising demand for Perkins engines, and in 1950 plans were drawn up for doubling the factory area. This second project is now approaching completion. It is noteworthy that although the factory area has been doubled, the output potential has increased fourfold. There has also been a very marked reduction in the man-hours expended per engine.

It is not intended that these notes shall deal in detail with all the production equipment nor with all the production methods, since in many respects the equipment and methods are broadly similar to those employed by other makers of high quality internal combustion engines. We may, however, note that some 75 per cent of the machine tools in use in the factory are less than three years old.

Before any reference is made to the actual production processes, it is advisable to describe in some detail the general layout of the factory and the various materials handling methods that have been adopted. This is because these factors play a very important part in the overall efficiency of the plant.

The original production factory, built in 1946/7, is a building 600 ft long × 200 ft wide. An extension 245 ft wide has since been added. This extension gives an added production area also 600 ft long × 200 ft wide. The remaining 45 ft of the extension are accounted for by a central roadway 20 ft wide and by space for the factory heating units, which were previously outside the wall of the original block.

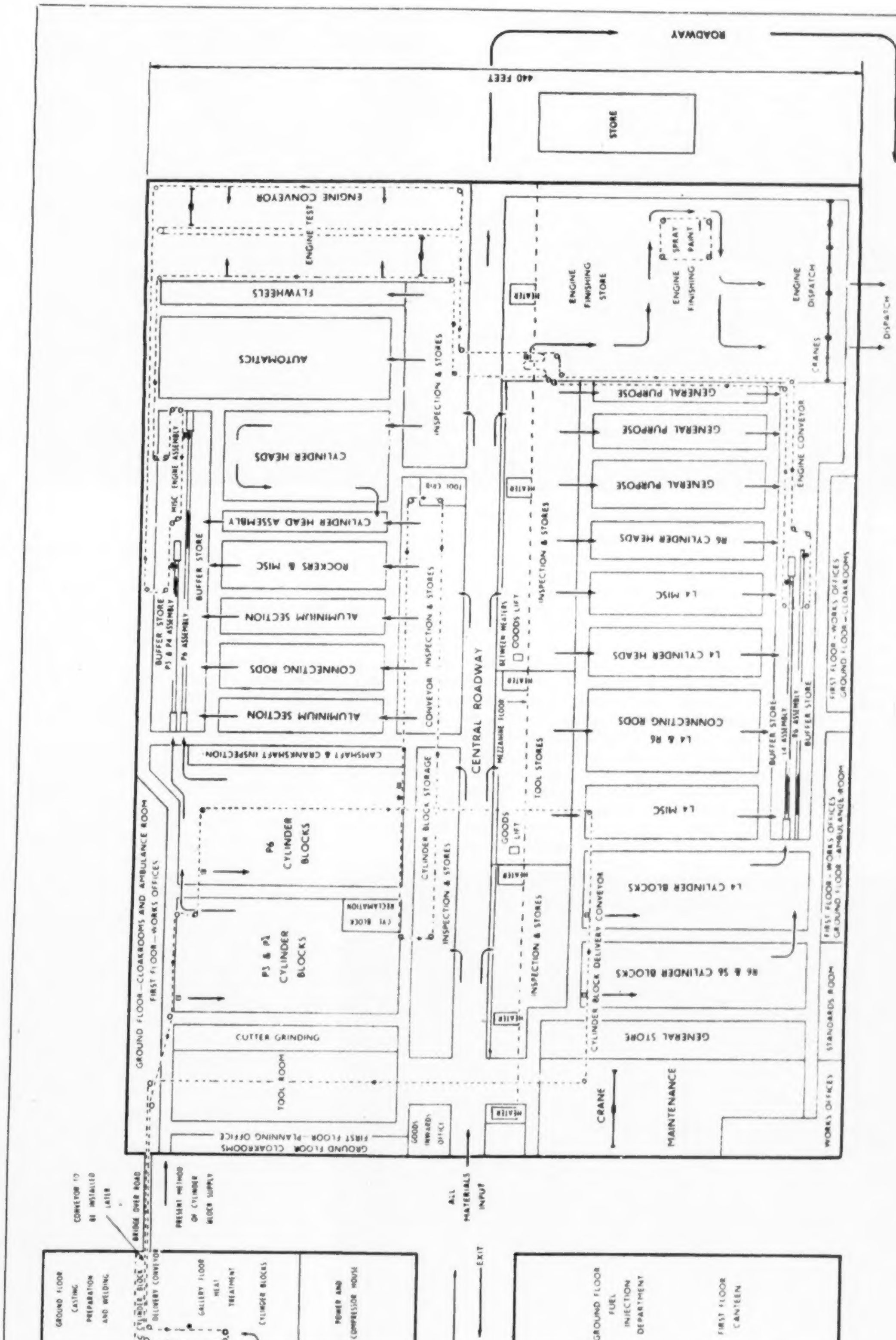
Six basic types of engines are produced. They are the three, four and six-cylinder "P" series engines, which are produced in the new block, and the six-cylinder "R" and "S" and the four-cylinder "L" types, which are produced in the original part of the factory. The complete factory layout and directions of work flow are shown in one of the illustrations.

Materials handling

As is only to be expected, one of the main aims in the planning of the factory layout was to reduce materials handling to a minimum. The provision of a central roadway has been a great help towards this desirable condition. Rough castings, such as those for cylinder blocks and heads, and rough stampings, such as those for connecting rods, are received in a separate building for inspection and any necessary fettling before they are delivered to the production process sections. All other bought-out goods are delivered direct to the appropriate point at one side or the other of the central roadway. Items that are common to all engines are delivered to a general stores for subsequent issue to the appropriate production point. Items that are peculiar to one type of engine are unshipped at the most convenient length along the roadway for issue direct to the process line.

As far as possible, bought-out

Part of the stores area adjacent to the central roadway



The new production factory of Frank Perkins Ltd., Peterborough

A general view of the machining section for P3 and P4 cylinder blocks

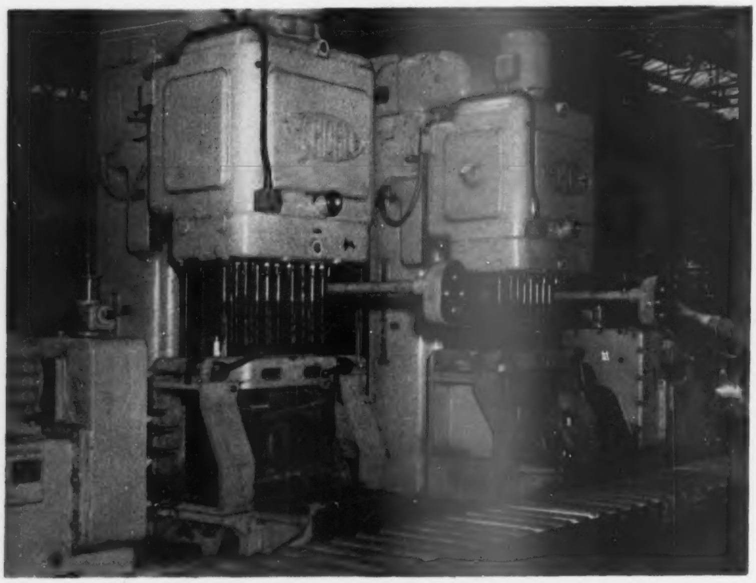
supplies are delivered in pallets. Frank Perkins Limited intend to make as wide a use as practicable of palletization, and they are already receiving co-operation in this matter from several suppliers. It was also decided to standardize the pallets in the greatest possible degree. The types of components were analyzed in relation to the process sections and, after full consideration of vehicle capacities for economical inter-works transport, a standard range of pallets was developed. The sizes decided upon are: —

48 in long, 36 in wide, 24 in deep with solid bottom and 2 in square mesh sides.

36 in long, 24 in wide, 24 in deep, solid throughout.

24 in square, 18 in deep, corrugated solid pallets.

Standard foot clearances have been allowed for lifting by fork truck, and normal "cup" locators are fitted to the base of each corner post. To provide additional means for movement, lifting eyes are provided in the tops of the corner posts. Each of the two larger pallets will take loads up to one ton; the maximum load for the smaller pallet is ½ ton. Standard flat pallets, both wood and steel, in two sizes, 40 in × 40 in and 36 in × 30 in, are also widely used.



Two Archdale two-way multi-drilling machines. One machine drills the head and front faces of the P4 block, but only the front face on the P3 block. The second machine drills the head and rear faces of the P3 but only the rear end of the P4 block

Special pallets have been designed for certain components, such as camshafts and copper water manifolds. Small parts, such as nuts and bolts, are issued to the production sections in metal tote boxes 15 in long, 10 in wide and 5 in deep. They are fitted with drop handles at each end and are housed in special support racks and shelves in positions convenient for the use of the parts.

Conveyancer and I.T.D. 5/6,000 lb capacity fork trucks with Perkins diesel engines are used for unloading materials along the central roadway and for general handling round the factory. Smaller electrically-powered models of the same make and Lansing-Bagnall pallet transporters are used for movement of materials along the process lines. For movement of materials over longer distances, petrol or electrically driven platform vehicles and tractor-trailer units are used.

Overhead conveyors installed by Geo. W. King, Ltd., are widely used for transport between sections. When the project is completed, cylinder block castings will be transported by overhead conveyor from the casting preparation shop to the appropriate process machining line. To conserve floor area for actual productive purposes another overhead conveyor carries one day's supply of machined cylinder blocks; in effect, this conveyor is the storage area for the buffer stocks. Assembled engines are transferred to the test shop on another overhead conveyor. Overhead conveyors will also be used for carrying engines from the test shop to the engine finishing section and thence through the spray booth.

Finally, careful thought has been given to the methods of materials handling on the process lines, a factor that is sometimes overlooked even in organizations that otherwise employ

excellent materials handling methods for movements between departments and sections. On the cylinder block and head lines, transfer of work from one machine to the next is generally by means of a roller track. Where necessary air hoists and grabs are installed for lifting the work from the track to the machine.

As far as practicable, the machines are arranged to have a standard loading height and the work-holding fixtures are arranged for through loading. Although the advantage of having a constant loading height for all machines is obvious, there are other factors to consider. For example, to maintain a constant and convenient loading height certain machines would need to be set in pits sunk in the floor. If this were adopted, the machine layout would be virtually inflexible. After due consideration of all the relevant factors it was decided that on certain process lines the machines should be mounted direct on the floor, irrespective of loading height, so that if in future a change in layout appeared desirable, it could be easily effected. To meet higher than standard loading heights on the cylinder block lines, short lengths of roller track with pneumatic lifting arrangements are installed at the appropriate stations. For smaller components, such as main bearing caps, elevator conveyors are used where the working station of a machine is considerably above the standard loading height.

In processing the cylinder blocks there is inescapable necessity to turn the component through 90 or 180 deg about its longitudinal axis from time to time. To allow this to be carried out easily without fatigue to the operator or damage to the work, roll-over devices are incorporated in the roller track.

Production layout

Practically all the productive processes are carried out in the main building. In fact, the only components produced in the auxiliary buildings are the various pipes and the injection nozzles. These notes will deal only with the processes carried out in the main building; nozzle manufacture will be the subject of a separate article.

The factory is bisected longitudinally by the central roadway so that there are two production blocks of equal area. "R," "S" and "L" engines are produced in one block and "P" series engines in the other. There are several sections in each block devoted to the production of only one specific component for a certain type of engine; in other areas work is carried out for all types of engines. For example all automatic machining is carried out in one section.

Although manufacture is restricted to a range of six basic engines, the output goes to some hundreds of organizations in 118 different countries; it is, therefore, not surprising that there are detail differences between engines of the same basic types that are intended for different applications. To deal with special requirements to suit particular applications there are three general purpose machining sections. These are essentially small quantity production sections.

The main production sections are laid out with the work flow at right angles to the central roadway. Every section is so laid out that the final machining operation is carried out as near as possible to the point at which the component will be needed in the sub-assembly or main engine assembly line. Although the machines are compactly arranged so that in many cases one operator can tend several machines, the operators have ample space. There are four engine assembly lines, two running down one side of the factory and two down the other, parallel to the central roadway.

Archdale horizontal multi-drilling machine with a three-station fixture that incorporates a manually operated pawl type transfer bar

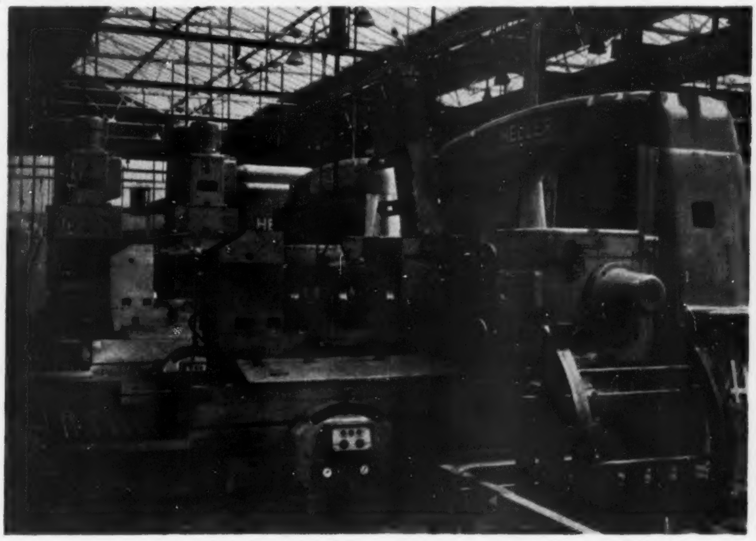
Cylinder block production

In some respects the most interesting of the cylinder block lines is that on which both P3 and P4 blocks are machined, since it has been laid out to allow a rapid change-over from one type to the other. Most of the machines in the line are used on both three and four-cylinder blocks, but for certain operations it has been advisable to have separate machines for each type. Change-over can be effected without stopping production for more than 20 minutes.

A Heller duplex mill with retractable heads is used for the first operation on both blocks. The work fixture has two stations, one for four-cylinder and one for three-cylinder blocks, so that to make the change-over from one to the other it is only necessary to adjust the table stops. At this operation the sump and head faces are rough milled.

The casting is loaded into the fixture with the off side uppermost. It is supported on two pegs which pass through holes cored to the water passages and contact two cylinder walls. A third support engages a flat pad on the sump flange. Air clamping is employed. It is so arranged that when the control lever is moved one pair of clamps push the casting to the left until the sump face contacts a pair of stop pads, while at the same time a pneumatic plunger pushes the work longitudinally until the rear end face contacts a location stop.

Two Wimet cutters are used. One is 16 in diameter and has 38 teeth and



Heller duplex mill for rough machining the head and sump faces of the L4 cylinder block

A Heller duplex machine for milling the ends of L4 cylinder blocks. In the background there are two Burkhardt and Weber horizontal multi-spindle drilling machines

machines the sump face; the other, for the head face, is 14 in diameter and has 34 teeth. Both cutters run at 63 r.p.m., the cutting feed rate is 20 in per minute and the depth of cut approximately ⅛ in. At the end of the forward stroke, the cutter heads are automatically retracted and the table returns to the forward position at rapid traverse. The floor-to-floor time for this operation is 3·2 minutes. A similar machine with a similar set-up is used for the next operation at which the sump face is finish milled and the head face semi-finished. The same size cutters are used as those used at the first operation and the feed rate is also the same. The milled sump and head faces give accurate height locations for subsequent operations.

At the next operation, which is performed on a Kitchen and Wade radial arm drill, holes are drilled in both the head and sump faces. Two of the holes in the sump face are also reamed to act as register points for accurate longitudinal and transverse location at subsequent operations. A trunnion type two-position indexing fixture is mounted on the machine table. The casting is loaded into the fixture with the sump face down. Transverse location is effected by two pegs that contact the side walls of two cylinders and longitudinally by a pin that contacts the rear end cylinder wall. The pegs for locating against the side walls of the cylinder bores are carried in a yoke in holes appropriate to the type of block that is to be machined. Loose jig plates are used for governing the positions of the holes drilled in the head face; bushed holes in the base of the fixture control the positions of the holes in the sump face.

From the radial arm machine, the casting is transferred to a Heller machine for rough boring the cylinder bores. At this stage separate machines are used for P3 and P4 blocks. There are two branches of roller track, one serving a six-spindle Heller vertical borer on which the P3 block is rough bored under the first three spindles and then semi-finish bored under the other three. The other track leads to two four-spindle Heller vertical borers on which the P4 block is rough and then semi-finish bored. Apart from the differences to suit the number of cylinders, the tooling and the work fixtures are basically the same for both P3 and P4 blocks.

The casting is loaded into the fixture with the sump face down and is located by manually-operated plungers that register in the sump face dowel holes. Snout type spindles are used. Each carries three inserted carbide cutters. The surface cutting speed is 200 ft/min, and the machine cycle is fully automatic. For both roughing and semi-finishing the feed rate is 2 in/min.

Following the semi-finishing operation in the cylinder bores the block returns to a common track for transfer to another Heller duplex mill with retractable heads, which is tooled for milling certain pads on both sides of the casting. The work is secured in the fixture by means of two pneumatically-operated clamps so placed that they will hold either a three or a four-cylinder block without any adjustment. Two Wimet cutters are used. Each is 6 in diameter and has 14 teeth. The surface cutting speed is 250 ft/min and the feed is 20 in/min.

Certain pads on the near side of the block still remain to be machined at the next operation, which is carried out on a Cincinnati plain mill. This machine has a higher working height than is standard for the line, and to allow easy loading and unloading a short length of the track is arranged for pneumatic elevation to the desired height.

Between the Cincinnati machine and the next operation, a roll-over fixture is incorporated in the track. It is used to turn the block through 180 deg to bring the sump face uppermost. In this position the casting is loaded into an Archdale limited-adjustment, multi-spindle drilling machine. On this machine the bearing cap stud holes are drilled and holes are drilled and countersunk in the sump face by means of stepped drills. There is a separate machine for each type of block.

The next machine, also a limited-adjustment Archdale vertical multi-drill, is used for both P3 and P4 blocks. It has 16 spindles and is arranged for rough and finish counter-

Burkhardt and Weber three-station transfer machine tooled for drilling, core drilling and reaming eight tappet holes in the L4 block

boring the bearing cap stud holes. The work fixture on this machine is interesting inasmuch as it incorporates a manually-operated transfer bar mechanism for indexing the work three times.

Transfer arrangements

A table which can be raised and lowered is incorporated in the fixture. The work is loaded with the table in the lowered position. The table is then raised to bring the sump face into contact with a datum face at the top of the fixture and locating plungers in the top of the fixture register in the sump face dowel holes. In this position six holes are rough counterbored when a P4 block is being machined.

When the machine head has retracted, the work is lowered and the transfer mechanism is actuated by handwheel to carry the block to the next station. At the predetermined position a positive stop prevents further rotation of the handwheel. The work is then raised and the six holes are finish counterbored. At the position reached after the second indexing, the other four holes are counterbored; they are reamed at the final position.

To ensure correct positioning, the transfer bar is operated in conjunction with a selector lever and a graduated index scale. The scale is so graduated and the locating plungers are so arranged that either P3 or P4 blocks may be indexed and accurately located through the successive positions in the fixture.

Before the next operation the block is turned on to the sump face in a roll-over fixture and is then transferred to an elevating fixture for loading in to a Cincinnati duplex mill that is tooled for machining the ends of the block. The milling cutters are 16 in diameter and have 24 teeth. They run at a surface cutting speed of 220 ft/min at a feed of 10 in/min and remove $\frac{1}{8}$ in to $\frac{3}{16}$ in of metal from each end face.

Huller two-station automatic transfer machine for drilling oil holes for the main and camshaft bearings

There are interesting features in connection with the next two machines in the line. They are both Archdale machines with two-way multi-spindle drill-heads, one vertical and one horizontal. For P4 blocks the first of these machines is used to drill 19 holes in the head face and 17 holes in the front end face, but for P3 blocks only the horizontal head is used to drill 17 holes in the front end face. On the second machine only the horizontal head is used for P4 blocks; it drills 14 holes in the rear end face. Both heads are used on P3 blocks, the vertical to drill holes in the head face and the horizontal to drill holes in the rear end face.

On the second machine, one of the locating plungers has an eccentric mounting in the fixture and can be moved into or out of the working position to ensure that when the vertical head is in use only P3 blocks can be loaded. Alternatively, when the vertical head is out of use and the plunger is in the other position only P4 blocks can be loaded.

At the next operation a horizontal drilling machine is used to drill the pressure rail in two stages. This hole is first drilled halfway through from one end and the casting is then reversed to allow the hole to be drilled through from the other end. The work fixture incorporates means for ensuring exact alignment. This operation has the longest cycle time in the machining sequence, the floor-to-floor time is 6·4 minutes.

For the next operation it is necessary to turn the casting to rest on the front end face. To allow this to be easily effected there is a special air-operated fixture at the junction of two lengths of roller track that are at right angles. Essentially, it is a trunnion mounted cradle that can be rotated about its horizontal axis, while the trunnion support can be rotated about a vertical axis. The casting is up-ended in this device and turned to face the work station on a Kitchen and Wade machine to which it is transferred by means of an air hoist. The machine is tooled for machining two starter cradle mounting pads. Two cutters, one for each pad, are mounted on the boring bar. Since the cut is intermittent a 12 in flywheel is mounted between the cutters to ensure a good quality of surface finish and long tool life.

At the completion of this operation the casting is returned to the trunnion

Archdale multi-spindle drill with a four-station fixture for drilling and rough and finish counterboring the bolt holes in cluster castings for P6 bearing caps

cradle, is turned through 90 deg and transferred to the roller track. Before it reaches the next machine, the casting is turned in a roll-over fixture on to its off side ready for loading into a Cincinnati plain milling machine tooled for gang milling the main bearing faces. All faces except the outer ones of the front and rear bearings are finished at this operation. A surface cutting speed of 200 ft/min and a feed of $1\frac{1}{2}$ in/min are employed.

An Archdale limited adjustment horizontal multi-spindle drilling machine is used for the next operation. The work fixture incorporates manually-operated pawl-type transfer bars by which the work can be indexed to three successive stations. At each position the work is accurately aligned with the locating plungers. This machine is used for drilling, reaming and countersinking holes in the cam side of the block. All three stations are, of course, loaded simultaneously. The machine for the next operation is of the same type but is tooled for drilling holes in the near side of the block.

To complete the drilling of holes in the ends of the block and to spot-face bosses on the sides, the casting is transferred by air hoist to an indexing fixture on a Kitchen and Wade radial arm drill. The fixture, which is manually indexed, allows the sides and ends to be brought into position quickly and easily. At three successive stations the holes in the sump and head faces, sides and ends are tapped before the bearing caps are fitted.

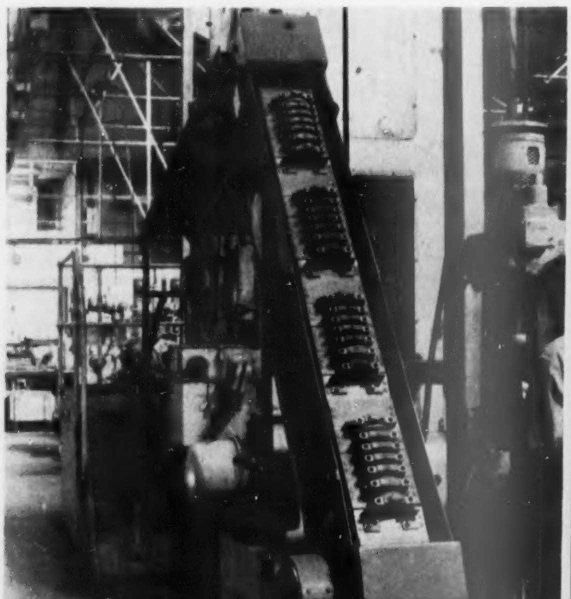
After the bearing caps have been fittted the block is transferred to an Archdale two-head horizontal borer tooled for core drilling the camshaft and main bearings and for machining the chain tensioner recess. One head is tooled for core drilling two camshaft bores and three main bearings from the rear end of the block. The other head carries a core drill for the remaining camshaft bore, a core drill for two main bearings in the case of a P4 block, and a four-tooth facing cutter for machining the chain tensioner recess. Both heads have variable feed motions so that when the tools clear one bearing the traverse changes from feed rate to rapid rate and then slows down as the next bearing is reached.

Semi-finish boring of the camshaft and main bearing bores is then carried out on an Asquith horizontal pull-boring machine. One boring bar carries three tools, one for each camshaft bearing; the other carries five tools for P4 and four tools for P3 blocks, one for each main bearing. The work fixture includes arrangements to allow the bars to be passed through the bores without fouling the casting before the operation and to allow bar withdrawal when the operation is completed.

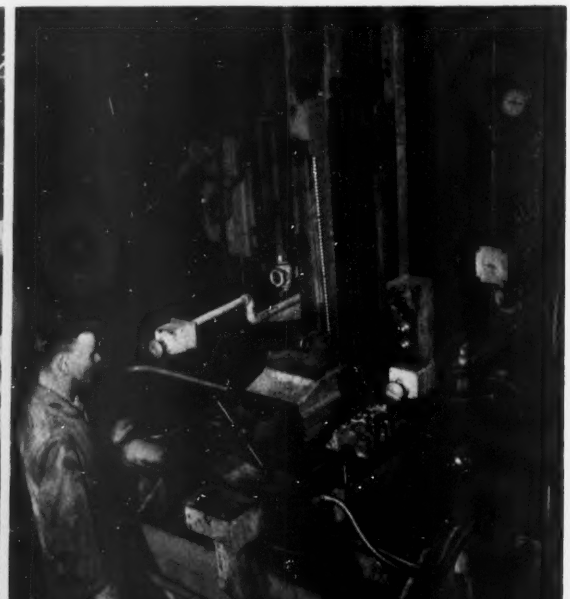
Briefly, the arrangement is that the work is pushed back in the fixture until it contacts a pair of stop pads. At this position it also depresses a switch that operates a green signal light to indicate that the work is in the correct position to allow the boring bars to be advanced to the starting position. When the bars are fully advanced, the back stops are freed pneumatically and the work is pushed by hand to the final position to allow the locating plungers to register in the dowel holes in the sump face. There are electro-mechanical interlocks to ensure that the machine cannot be started until the bores and the cutters are correctly aligned. At the completion of the cut, the boring bars must be rotated manually to bring the tools into the correct position for withdrawal and the work must be moved forward before the rapid withdrawal motion can be engaged.

The main bearing bores are then finished to size on a Heald Borematic. A quickly-detachable boring bar with a tool for each bearing is used. When the casting is placed in the fixture it rests on six spring-loaded plungers and is in such a position that the boring bar can be passed through the bearings without the tools fouling the work. As soon as the bar is in position, the plungers are lowered and boring starts. At the end of the cut, the spring-loaded plungers are raised and the work is lifted sufficiently to allow the bar to be freely withdrawn.

From the Heald machine the casting is transferred to a Heller duplex mill on which the front and rear faces are finish machined. The set-up is approximately similar to that used for rough machining the ends. The bearing caps



An elevator conveyor taking bearing cap cluster castings to a broaching machine



Weatherley Oilgear surface broaching machine for splitting cluster castings into individual caps

are then removed and the camshaft bearings are fine bored on a Heald double-end Borematic. The front and intermediate bearings are machined from one head and the rear bearing from the other. Oil holes in the main bearings are then drilled and counter-bored on a Kitchen and Wade radial arm machine. Following this, the head face is given a finishing skin cut on a Heller vertical mill. A 14 in diameter, 34-tooth cutter is used at a surface cutting speed of 300 ft/min, and a feed of 20 in/min.

At this stage there are separate machines for P3 and P4 blocks for fine boring the cylinder bores. They are Heller machines of the types described earlier for rough and semi-finishing the bores. When the fine boring is completed, a recess for locating the cylinder liner is machined in each bore on a Pollard column drilling machine. The liners are then pressed in and the block is transferred to a Heller multi-spindle vertical borer for fine boring the liners to a total tolerance of 0·001 in. A rough honing operation is then carried out on a Kitchen and Wade single spindle machine. The stones are 200/260 grit and run at 20 r.p.m. and 80 strokes/min.

There is an appreciable length of track before the next machine is reached. This allows the honing fluid to drain from the block and also allows frazing and certain grinding operations with hand tools to be carried out. Core plugs are then fitted and the block is pressure tested, water jackets at 50 and oil-ways at 80 lb/in^2.

From the water test the block returns to the roller track and a full inspection is carried out. There is an appreciable length of track before the next machine is reached. The buffer stock carried on this length ensures a minimum settling time of 16 hours before the finish honing is carried out on a Kitchen and Wade single-spindle machine.

From the finish honing operation the casting passes to a length of track on which it remains for a period to allow the honing fluid to drain away. It is then transferred by pneumatic hoist to a sub-assembly section. The casting passes through this section on a roller track, and to ensure that the joint face will not be damaged, it rests on a small pallet. Main bearing shells, dowels and thimbles are fitted and the sub-assembly is passed to a Heald Borematic on which the shells are fine bored.

From the boring machine the block is transferred to another length of roller track, where the main bearings and caps are removed and placed in a tray to accompany the casting to the engine assembly line. All the studs are then assembled and the oil seals are fitted to the rear bearing. The block then passes through a Curran washing machine, and after it leaves the machine all passages are blown out by means of a fine nozzle fitted to an air line. To complete the work preparatory to transferring the block to the engine assembly line, the crankshaft is fitted.

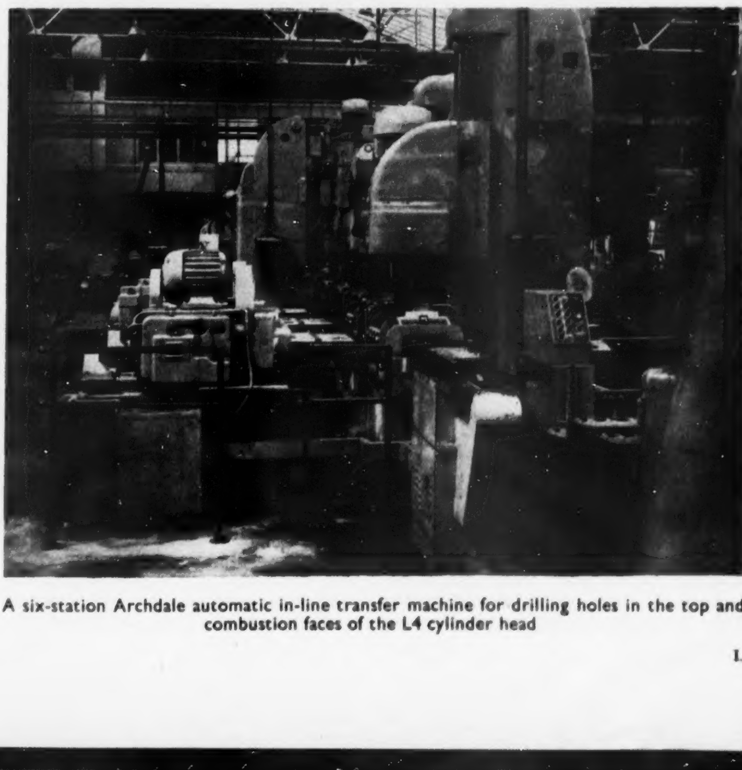
Asquith three-way drilling machine for rough, semi-finish and finish machining the combustion chambers in P6 cylinder heads

L4 cylinder blocks

Cylinder blocks for L4 engines are produced on a more recently installed line. The operation sequence is broadly the same as that for P3 and P4 blocks but there is considerably less handling to be done. With few exceptions the machine tools are of German manufacture, a development that was forced by the difficulty in obtaining delivery of British machine tools.

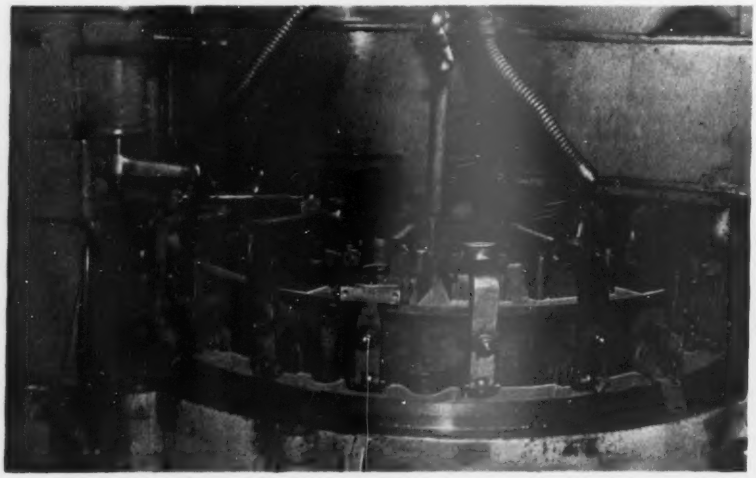
As with the P3 and P4 blocks, the machining sequence starts on a Heller duplex mill with retracting heads for rough milling the head and sub-faces, and at the second operation a similar machine finish mills the sub-face and semi-finishes the joint face. For the third operation, instead of a radial arm machine, there is a Burkhardt and Weber multi-spindle vertical drill. It is used to drill the holes in the sump face and to ream the dowel holes for locations at subsequent operations. This machine has two heads and there is provision for direct transfer from one head to the other.

Further savings in handling in comparison with the P3 and P4 blocks are effected by the greater use of multi-spindle drilling machines and an even more marked saving comes through the



A six-station Archdale automatic in-line transfer machine for drilling holes in the top and combustion faces of the L4 cylinder head

The second Archdale automatic transfer machine for L4 heads. Finishing operations on valve guide holes and valve seats are carried out on this machine



The loading station of the Blanchard automatic grinder for connecting rods and caps

use of multi-spindle tapping machines. A greatly improved method is also used for drilling the oil pressure rail. For this operation on the L4 block, a double-ended multi-head Burkhardt and Weber machine is used. There are three stations. At the first station the oil pressure rail is drilled 4 in in from each end and from one head the idler hole is drilled through; at the second the oil pressure rail is drilled to a depth of 8 in from each end and the idler hole is opened out to a depth of $1\frac{1}{16}$ in; finally at the third station the pressure rail is drilled through and the idler hole is recessed and chamfered.

Another interesting Burkhardt and Weber machine is a vertical multi-spindle drill with a three-station fixture arranged with hydraulically-operated transfer of the part from station to station. At the first station eight tappet holes are drilled; at the second they are core drilled and reamed at the third station.

At a later stage a two-station Burkhardt and Weber machine is used for drilling and fine boring the oil pump bore. At the first station the bore is drilled to two diameters with a stepped drill. The block is then moved by a manually-operated transfer bar to the second station where the two diameters are fine bored. Of the remaining operations on this block, the only one calling for comment is the set-up for drilling the oil holes from the main bearings into the pressure rail. A five-spindle Huller machine is used. It has five independent heads set at the appropriate angles. There is reciprocating motion for each spindle. A two-station fixture is used. Three holes are drilled with the block in one station and the work is then transferred to the second station where two holes are drilled.

Main bearing caps

Interesting and remarkably efficient methods are used for the production of main bearing caps. These components are made from Meehanite cluster castings; that is, a P6 integral casting eventually becomes seven caps of four different sizes, a P4 casting becomes five caps of four sizes and a P3 casting becomes four caps of three sizes. The production methods are substantially the same for the caps for all three engines, and in fact, many of the machines are used for all three. These notes will describe the methods employed in the production of P6 caps.

Machining starts on a Weatherley vertical surface single-slide broaching machine with a two-station fixture. It is tooled for broaching the joint face and the backs of the bolt bosses. At the first station, location is taken from the rough half bore and the bolt bosses are finish machined. At the second station, location is taken from the machined back of the bolt bosses and the joint face is broached in relation to the back of the bolt bosses. The broaching speed is 16 in/min.

The integral casting is then transferred to an Archdale 42-spindle vertical drill with a four-station fixture arranged for manual indexing. Loading is effected at the first station with location from the back of the bolt bosses and the half bore. At the second station the 14 bolt holes are drilled. They are counterbored and countersunk at the third station, and reamed at the fourth. Following this, five holes for suction pipe clips are drilled at one station on a Herbert two-spindle machine and then countersunk and tapped at the second station. A Herbert single-spindle drill is then used for countersinking the bolt holes on the back of the bolt bosses. For the final machining operation on the integral casting, the work is transferred on an elevating conveyor to a Weatherley Oilgear vertical surface broaching machine for parting off into individual caps. At the same time side-cutting broaches machine the ends of the casting. Five passes are necessary to complete this operation. The cycle is fully automatic with the shuttle table advancing a predetermined amount towards the tools between successive strokes of the slide and then retracing to the loading position after the last stroke.

Only a single operation is needed to complete the intermediate bearing caps; a dowel hole is drilled and reamed. Front main bearing caps are machined in pairs on a Drummond capstan lathe. On this machine the two caps are faced and a diameter is rough and finish turned from the front tool box, while a groove is formed from the rear tool post. Three angular holes are then drilled through to the half bore on an Archdale radial drill. Following this, front and rear caps are loaded in pairs into a drilling fixture on a Herbert two-spindle drill. Four holes are drilled and the pair of components is then transferred complete to a tappet fixture under the second spindle where the

holes are tapped. To complete the machining, a dowel hole is drilled and reamed on an Archdale single spindle machine. The sequence for the rear main bearing cap differs only slightly from that for the front cap.

Cylinder heads

It is not intended to follow the machining sequence on any of the cylinder heads in detail, but to refer only to some of the more interesting developments. For example, there is an Asquith three-way horizontal screw-unit multi-drilling machine with a manually indexed table for roughing, semi-finishing and finishing the combustion chambers in the P6 cylinder head. The components rest on the joint face and are located in the work fixture from the tappet holes. Combination form cutters mounted at the first working station rough form the combustion chambers and recesses. At the second working station the combustion chambers are finished and the recesses are semi-finished. The recesses are finish machined at the final station. All the tools work at a surface cutting speed of 40ft/min and a feed of 1 in/min.

Probably the most highly developed line for cylinder head production is that for the L4 component. This line includes two Archdale multi-station in-line automatic transfer machines. One interesting set-up in this transfer line is on a Heller duplex milling machine. The work fixture on this machine is arranged to hold three cylinder heads, two with their longitudinal axis parallel to the longitudinal axis of the machine table and the other with its longitudinal axis at right angles. Each casting passes through the machine three times. At the first pass both ends are machined; the casting is then transferred to the second position in the fixture and one side is machined at the next pass; the casting is then transferred to the third position for the other side to be machined at the third pass.

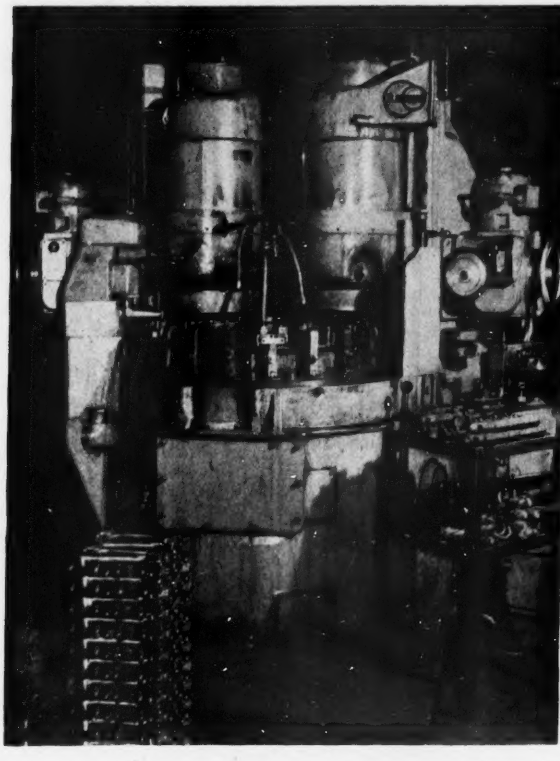
Each of the Archdale automatic in-line transfer machines has six working stations. The component is loaded into the first machine with the top face uppermost. At the first station 18 holes are drilled in the top face from a vertical head, and four spherical combustion chambers are rough bored and rough recessed from a horizontal head. At the second station, eight valve guide holes and four injection holes are drilled in the top face and the spherical combustion chambers drilled at the first station are finish machined and the recesses are semi-finished. Four injection holes are drilled in the top face and six recesses in the combustion face are finished at the third station. At the fourth station 10 holes are drilled in the top face and two are drilled in the exhaust face. The four injection holes in the top face are drilled through at the fifth station and eight holes are drilled in the combustion face. At the final station eight valve guide holes are semi-finish reamed and four injection holes are finish bored in the top face, and eight part-drilled holes in the combustion face are drilled through.

The second transfer machine follows immediately in the sequence. For loading into this machine, the casting is turned to bring the joint face uppermost. At the first station, four inlet and four exhaust valve ports are core drilled in the top face and four combustion throats are core drilled in the combustion face. The valve seats and recesses are cut from one head and the spray clearance for the nozzle apertures is cut from another head at the second station. There is only a single head at each of the remaining stations. At the third station four combustion throats are core drilled and are finished at the fourth station. The remaining combustion throats are finish formed at the fifth station. To complete the operations on this transfer machine the inlet and exhaust valve seats and ports are finish machined and the valve guide holes are finish reamed.

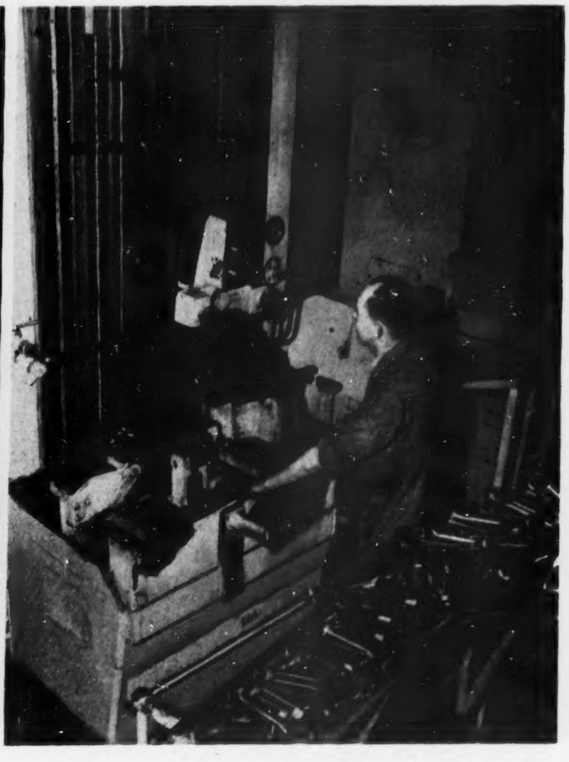
Connecting rods

The connecting rods are received as separate forgings. They are in chrome-molybdenum steel. Certain preparatory work is done on each in the casting preparation department before they are delivered to the process machining line. These notes will deal only with the production of connecting rods for the "P" series engines. The rods produced in this section are common to all three engines.

Before the machining proper starts



Blanchard automatic rotary surface grinder for connecting rods and caps



Weatherley Oilgear double-slide machine for broaching 'P' series connecting rods and caps

Burkhardt and Weber 32-spindle machine for drilling and reaming bolt holes in 'P' series connecting rods and caps

on the rod, the large end is centre drilled on the joint faces at two points coinciding with the bolt hole centres, and the small end is centre drilled in an extension boss. These centres act as location points for subsequent operations. They are of such importance as to merit 100 per cent inspection. The rod is then mounted in a Herbert Carbicut lathe for turning the piston bore clearance at high speed.

At the next operation the side faces of rods and caps are ground together on a Blanchard rotary automatic surface grinder. This is a two-head machine with one head set slightly lower than the other so that rough and finish grinding are effected in one pass. The caps and rods are mounted alternately in fixtures on the rotary table. These fixtures are arranged in pairs, one pair is used for the first pass, after which the components are turned and mounted in the second pair, which are slightly higher, for grinding the other side.

Electronic sizing

The height of the wheels in relation to the work is controlled by an electronic sizing device. Each piece as it leaves the finishing wheel passes under a feeler that registers wheel wear from the increased thickness of the component. When the tolerance limit is approached, the wheel heads are automatically fed down a predetermined amount. The work holding fixtures incorporate clamps which are automatically operated by the action of a cam track as the work approaches the first wheel. They are automatically released after the work leaves the second wheel.

For the next operation the rod is located from the three centres in a

The assembly conveyors for Perkins 'P' series engines

four-station fixture on a Burkhardt and Weber nine-spindle machine. Three rods are mounted at each station. At the first working station the small end of each rod is drilled to 1 in diameter, at the second the hole is core drilled, and at the third reamed to size. The machine cycle is completely automatic and the components are loaded and unloaded during the cutting cycle.

Location is again taken from the three centres for the next operation, at which the small end bore is fine bored and chamfered. A four-spindle Precimax machine is employed. It has a four-station fixture so arranged that while two components are being machined the other stations in the fixture are being unloaded and re-loaded.

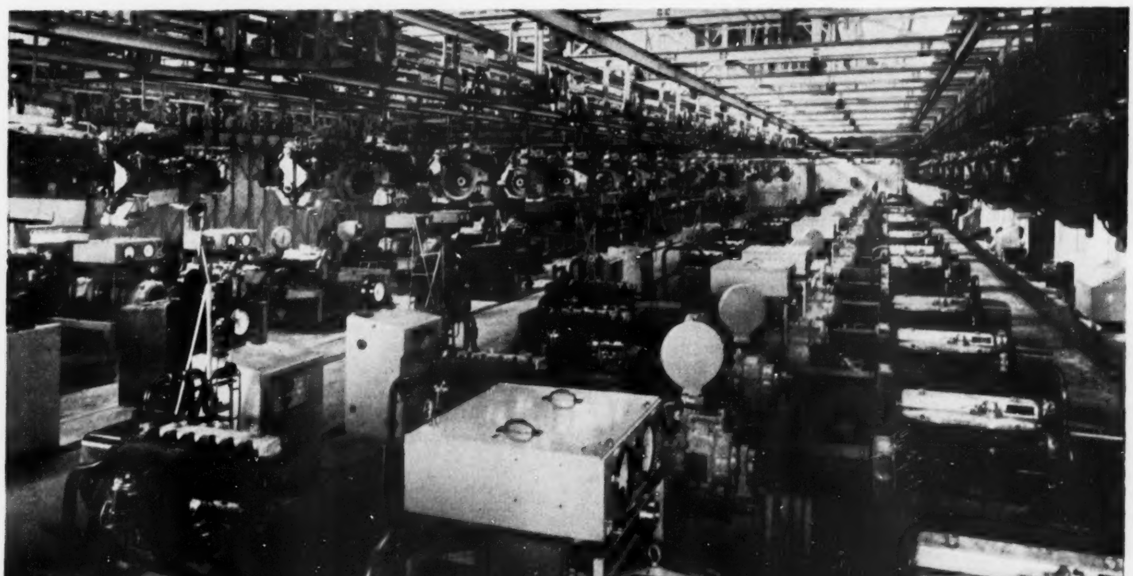
Rods and caps are machined simultaneously on the next machine in the sequence, a Weatherley Oilgear vertical double - slide surface broaching machine. There is a two-station fixture for each slide, one for rods and one for caps. At the first rod station the component is located from the small end bore and the piston bore clearance for broaching the half-bore. After the half-bore is broached the rod is transferred to the second station where the joint faces and the backs of the bolt bosses are broached. Clamping is effected automatically.

In the first cap fixture the component is located on one of the ground sides and the sides and backs of the bolt bosses are broached. At the second station the half-bore and the joint faces are broached. Two cutting speeds are employed. The first 53 in of slide travel is at 180 in/min; to give a good quality of surface finish this is slowed down to 60 in/min for the final 12 in.

Overhead turn-over fixture for changing the engine location in the assembly fixture

A 32-spindle Burkhardt and Weber machine is used for the next major operation, drilling and reaming the bolt holes. It has a five-station fixture arranged to take four components at each station. To ensure exact alignment of the bolt holes a cap and rod are assembled together in each work-holding fixture. At the first working station two holes are part drilled in each of the four assemblies. They are drilled through at the second working station, rough reamed at the third, and finish reamed at the fourth. From this stage on, the cap is kept with the same rod. This machine has a very high output rate, since not only are four rods and caps produced simultaneously but the speeds and feeds are also high. Drilling and rough reaming operations are carried out at 462 r.p.m., and the finish reaming at 180 r.p.m. The feed rate is 0·94 in/min.

A U.V.A. automatic internal grinder is employed for grinding the large end of the assembled rod. The location is taken from the small end bore and one side of the large end. Rough and finish boring are effected at one setting. The roughing operation is manually con-



The test shop in the new Perkins factory

trolled to leave about 0·008 in to be removed during the automatic finishing cycle. To complete the machining the small end bush is fine bored and the rods are then weighed into sets of six, four and three for P6, P4 and P3 engines. Each set must balance within 2 oz.

Assembly

In general, the sub-assembly line may be regarded as continuations of the machining line, with components passing direct from the final machine to assembly. The sub-assembly lines are arranged to feed direct to the main assembly conveyors. There are four engine assembly conveyors, one for P6, one for P3 and P4, one for L4 and one for R6 engines.

Three of these conveyors, the most recently installed, are of the pallet type with stations 4 ft 6 in apart. The fourth, for R6 engines, is of an older type with 7 ft 6 in station spacing. On this older conveyor the engines are assembled on trolleys attached to a floor conveyor. The trolleys are designed to allow the assembly to be rotated round the crankshaft axis so that any desired face can be brought to a convenient working position.

Pallet conveyors

On the other engine assembly conveyors, each pallet carries a turntable fixture that can be rotated through 360 deg about a vertical axis, so that the engine can be turned to any convenient position to allow work to be carried out from only one side of the conveyor. The transfer of cylinder blocks to the appropriate assembly conveyor is effected by means of a King electric hoist with a power traction unit. A special grab is used so that the block may be turned through 180 deg. Operation of a control switch causes the block to be transferred automatically to the assembly conveyor. Empty grabs are returned in a similar manner.

During assembly there is necessity to turn the engine about its crankshaft axis. At each station where the engine needs to be turned, there is a King electric hoist with a special turn-over grab. The turntable fixtures on the pallets have been designed to accommodate the engine in three different positions, that is, on the head face, the sump face and the side face.

The pallet fixtures on the assembly conveyor for P4 and P3 engines are not only designed to take the engine in three positions but also to take both engines. Change-over from one engine to another is effected by a simple adjustment of the supports. An interesting feature of the pallet type assembly conveyors is the arrangement made for returning the fixtures from the end of the track to the beginning. With the normal arrangement there would, of course, be as many fixtures on the return length of conveyor as on the working length, but this installation designed and built by Geo. W. King Ltd. has a fast return conveyor. At the end of the assembly line the pallet, and with it the empty fixture, is freed from the conveyor chain to run down a gradient on to the fast return conveyor. Because of this arrangement, only two spare fixtures are needed to keep the 37 working stations supplied.

Engine testing

Assembled engines are transferred to the test shop on a King overhead conveyor installation. This conveyor feeds switch lines in the test shop. These lines hold a buffer stock of engines between the assembly and the test beds. There are seven overhead travelling transporters for lowering engines on to the test beds. Every engine is given a functional test for performance and fuel consumption. The type of test brake mainly used is a Perkins-designed bed based on an electric motor generator that forms a load resistance against which the engine is run. Power thus produced is fed back into the mains and used in the factory. To economize in floor space the beds are constructed in pairs. There are 25 such units. Heenan dynamatic dynamometers are also used. As far as possible all the equipment in the test shop has been designed to be applicable to every type of Perkins diesel engine.

After the test is completed the basic engine is taken by overhead conveyor to the engine finishing line, where the special parts for the specific applications are fitted. Finally, the engine is spray painted to specification.

A remarkable increase both in total output and in output per man-hour that has been achieved in this new factory is the cumulative effect of many different factors. Some of the more important factors may be briefly recapitulated. In the first place, attention may be drawn to the very efficient use that has been made of floor area. At every station there is adequate room and yet the machines are so compactly arranged that where the time cycle allows, one operator is able to tend two or more machines. The policy of maintaining only small buffer stocks of components and sub-assemblies has also played an important part in ensuring that the maximum possible amount of the floor area is devoted to actual production processes. Finally, because of the very wide use of mechanical handling equipment, the non-productive/productive labour ratio is very low.

AERODYNAMIC STABILITY

A PAPER entitled "Wind Effects on Car Stability," by W. E. Lay and P. W. Lett, Jr., is published in an *S.A.E. Preprint*, March 3-5, 1953. In it, the effects of side winds and gusts on the stability of cars are examined, and the results obtained from road tests on a single car are discussed.

At average vehicle speeds, side winds produce resultant winds that vary in angle considerably. For stable performance, the tendency of the car to yaw should be minimized whatever the magnitude or direction of the side wind. The thrust due to wind on the body of a moving car is transmitted through the body mounts, etc., to the frame and hence to wheels and tyres. A new 1947 automobile was obtained for the tests, and a supporting system, which is described in detail, was devised to permit separation of the body and chassis, the body being carried on support columns to elevate it ⅛ in above its normal position on the chassis. Safety bolts were fitted; they allowed body movements only to the limit required for the tests. Wind forces acting on the body subjected the columns to bending strains that were measured with strain gauges. Instruments were also installed for the measurement of the speed and direction of the winds. Over 1,000 miles of road testing was carried out and the procedure adopted was aimed at isolating wind forces so far as possible. Test runs at various speeds were made in both directions over selected sections of road, and data were recorded photographically.

With the change from the head-on direction of the resultant wind, the side force on the car increased by an amount greater than the increase of cross-sectional area of the car normal to the wind. This effect was most marked when the resultant wind angles were within 10 deg of head-on. As wind direction shifted from head-on to the side-on direction, the centre of pressure of the resultant wind moved from a forward position towards the centre of gravity. Yawing tendency was most affected by small changes in resultant wind angle from a head-on direction. A gust at an angle to the path of travel of the vehicle caused a sudden lateral displacement, and required corrective action by the driver to regain the required course af travel. Aerodynamic stability depends primarily on: (1) to what degree lateral thrust forces from winds are equalized between front and rear wheels; (2) air resistance of the car to side winds. Fundamentally, body design is the key to the problem. (*M.I.R.A. Abstract No.* 6277.)

COMMERCIAL VEHICLE GEARBOXES

Notes on the Design and Production of David Brown Units

J. T. Riley, A.M.I.Mech.E.*

SINCE the end of the war the British commercial vehicle industry has made considerable and successful efforts to establish markets throughout the world. During the past five or six years there has been something like a threefold rise in the volume of commercial vehicle exports; this is a great tribute to the quality of the vehicles. A considerable part in achieving these results has been played by specialized manufacturers of important units.

For example, the extent to which proprietary gearboxes are used by British builders of commercial vehicles is perhaps not generally appreciated. This system of using proprietary gearboxes is widely used in the United States of America, where a large proportion of the gearbox units are supplied by three companies. Similarly, in this country, many of the leading manufacturers fit proprietary gearboxes manufactured by David Brown and Sons (Huddersfield) Ltd. These notes deal with the design of, and the production methods for, some current David Brown units.

This system leaves all the problems peculiar to gear and gearbox production to the gear specialist. It is claimed, with some justification, that it results in better and cheaper gearboxes. This is due to the concentration of effort which the gear specialist can bring to bear on his work, since his whole energy and outlook are devoted to one aspect only, gears and gearboxes. It is, of course, necessary to maintain the closest liaison with the vehicle builder so that there is full knowledge of all details that may influence design.

While a long list of customers is always gratifying, such a position is not without dangers for the gearbox manufacturer. The more numerous the customers, the greater the probability for a wider range and variety of units, and unless care and foresight are exercised in deciding just how far to go in meeting all demands, the point may be reached where economical production is no longer possible because of the endeavour to produce a wide range of boxes in small quantities. Regular personal contact with the vehicle builder provides the best means of reaching a decision on this very important aspect of production.

Design

The illustrations and descriptions which follow cover three different sizes and types of David Brown commercial vehicle gearboxes. They are typical of the considerable range manufactured by this Company. All incorporate layshafts, the most popular type in use in this country. They are designated 557, 542 and 437. The first figure in the designation denotes the number of speeds obtainable from the box, while the last two figures indicate the centre distance, which for model 557 is approximately 5·7 inches.

Model 557 gearbox, shown in Figs. 1 and 2, has five forward speeds and one reverse and is suitable for a maximum engine torque of 350 lb-ft. Complete with bell housing, it weighs approximately 470 lb. The aluminium case is made in halves, and the upper half can be removed without any disturbance of the internal mechanism. The proportions and dispositions of the ribs, together with the additional support provided by an intermediate wall carrying the centre bearings, combine to give exceptional stiffness and freedom from resonance.

All the gears are made from 3½ per cent nickel chrome case-hardened, which has been shown by experience to be the most suitable for withstanding the severe loading imposed in service. A direct top drive is employed. This substantially increases the life of the gearbox in comparison with similar units incorporating an overdrive top gear.

In this country it has been established that something like 75 per cent of the total running time is done in top gear, whether it be a direct drive or an overdrive gear. If a direct drive top gear is used, the torque is transmitted directly through the box without any of the gears being used, and it is considered that for service in this country this outweighs the advantages of an overdrive top gear.

It is, however, acknowledged that an overdrive top gear can prove useful under certain conditions. For example, where there are long stretches of level road, it is advantageous to maintain the desired road speed with the engine revolutions as low as possible; this can be effected by the use of a gearbox with an overdrive top gear. There is the disadvantage that when a vehicle is operating in the overdrive ratio gears, even a small gradient will probably necessitate a change into a lower gear.

All the constant mesh gears, that is, the layshaft driving gears and the clutch-operated third and fourth speed gears, of the 557 box are of the single helical type. The first and second speed gears are of the straight spur type with sliding engagement. All the gears are accurately generated, shaved after

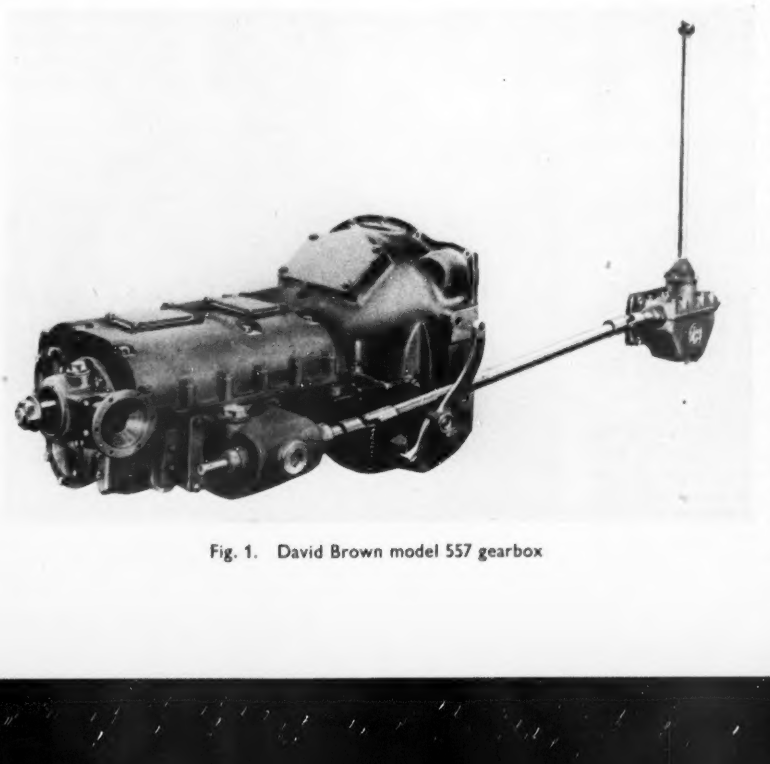


Fig. 1. David Brown model 557 gearbox

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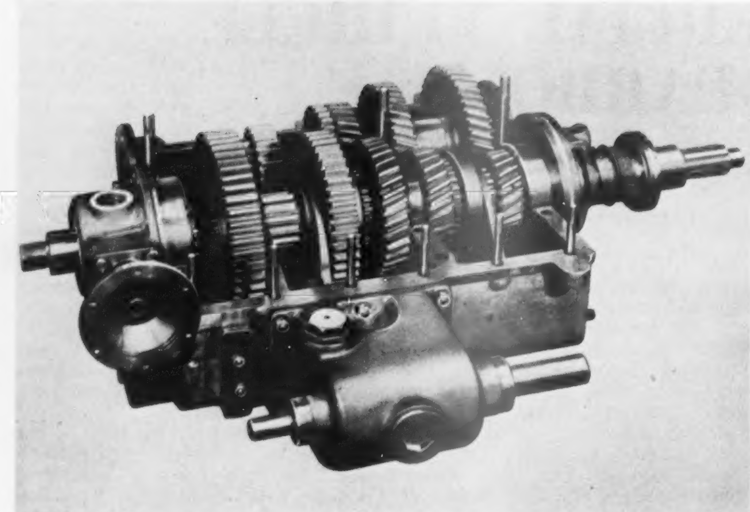


Fig. 2. Model 557 gearbox with top cover removed

cutting, and lapped after hardening. To engage the reverse gear, it is necessary to lift the change speed lever over a stop; this prevents the possibility of accidental engagement. The gear ratios are:—first, 7·92:1; second, 4·68:1; third, 2·74:1; fourth, 1·565:1; fifth, 1:1, and reverse, 7·92:1.

All shafts in the 557 gearbox are made of 3½ per cent nickel chrome steel, case-hardened and ground on the journals. The mainshaft is supported in two bearings and the layshaft on three. This arrangement allows the bearing spans to be kept as short as possible, and keeps the maximum number of gears close to the bearings. In a large degree, the quietness of the box in operation depends upon these desirable design features.

The intermediate wall in the case supports a centre bearing for the layshaft and provides independent support for the final shaft. This arrangement eliminates the need for a spigot bearing and is a great improvement, since the spigot bearing was a far from ideal arrangement acting between shafts running at different speeds and, in reverse, providing a common bearing between contra-rotating shafts.

The primary shaft is carried on a roller bearing and a location bearing is provided to take the axial load from the single helical gears. Three roller bearings support the layshaft, the two outer bearings having lips to locate the shaft axially. The front end of the mainshaft is carried on a roller bearing and on a deep groove ball bearing at the rear end. Roller bushes, each with two rows of large rollers, are fitted to each of the loose mainshaft gears. All bearings which are supported in the aluminium case are housed in steel liners. These effectively preclude any possibility of the outer races creeping. Oil filling, oil level and drain plugs of ample dimensions are positioned to give the greatest possible accessibility and the drain plug is fitted with a magnet to collect and retain any foreign matter of a ferrous nature.

An auxiliary drive with a suitably proportioned facing is provided to take a high pressure pump for brake operation. This is positioned at the rear of the gearbox so that the brakes are energized whenever the vehicle is moving, even though in neutral. The ratio of the speedometer drive gears, which are housed in the rear end cover, is suitable for the latest type of electric speedometer, which gives accurate readings, a comparatively low cable speed and, as a result, maximum life of the speedometer cable. Both the speedometer and auxiliary drive gears are carried on ball bearings.

Model 542

This gearbox, see Fig. 3, has five forward speeds and one reverse and is suitable for a maximum engine torque of 204 lb-ft. Its approximate weight complete with bell housing is 150 lb. It has proved very successful in use in conjunction with the Perkins P.6 engine. Generally the design is similar to the model 557 previously described. The first and second speed gears and the reverse gear are sliding gears with straight spur teeth; the remaining gears are in constant mesh and have single helical teeth. All the shafts are mounted on ball and roller bearings, the case is in halves to facilitate servicing, and this also allows a centre bearing to be incorporated on both main and layshafts. Two S.A.E. standard facings are provided for power take-off units.

The ratios are:—first, 8·14:1; second, 4·73:1; third, 2·79:1; fourth, 1·6:1; fifth, 1:1; reverse, 7·76:1. Alternative gear ratios can be supplied giving first, 6·61:1; second, 3·56:1; third, 1·88:1; fourth, 1:1; fifth, 0·812:1 and reverse 6·28:1. It will be seen that these ratios give overdrive conditions in the fifth gear that should only be used with light loads and easy running conditions.

The 542 box is designed complete with forward control for unit or amidships mounting, and in these applications the box is mounted horizontally to give maximum road clearance and minimum interference with the body space. An alternative design provides a special top cover arranged for centre control.

Model 437

This gearbox has four forward speeds and one reverse and is suitable for a maximum engine torque of 124 lb-ft. Its approximate weight complete with bell housing is 120 lb and it has proved very successful in use in conjunction with the Perkins P.4 engine. The ratios are:—first, 5·72:1; second, 3·17:1; third, 1·78:1; fourth, 1:1, and reverse 6·86:1.

The model 437 gearbox is suitable for either right-hand or left-hand control. It is shown in Fig. 4 with the change speed lever mounted in a central position, but remote control mechanism can be fitted if specially requested. A No. 3 S.A.E. bell housing is fitted as standard. The case is of cast iron, in one piece, with the selector lever housing acting as a top cover

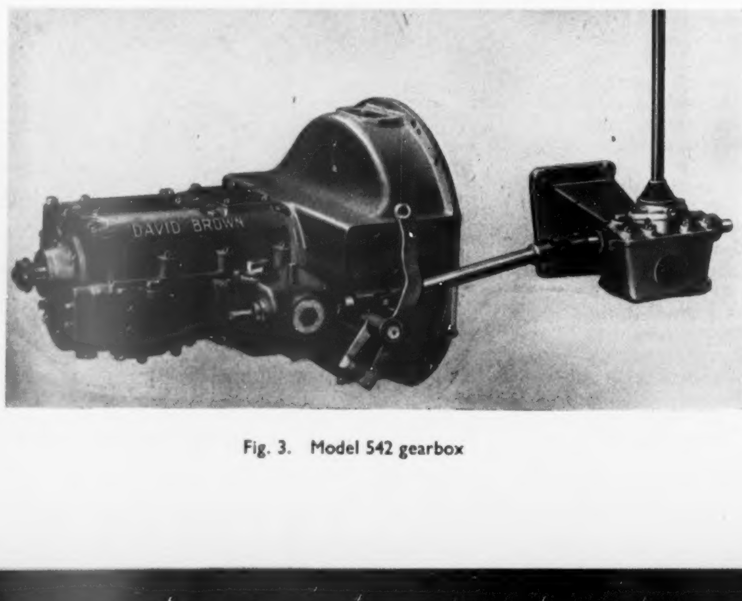


Fig. 3. Model 542 gearbox

removable as a unit for inspection purposes. All the gears are of 3½ per cent nickel chrome case-hardened steel. A direct drive top gear is employed. The layshaft driving gears and the second and third speed gears are in constant mesh and have single helical teeth, whilst the first speed and reverse gears have straight spur gear teeth and slide into engagement.

The primary shaft is carried on a ball location bearing designed to take both journal and thrust loads. A bearing of the same type is used to locate the mainshaft at the rear end. The forward end of the mainshaft is located on a roller spigot bearing, with the rollers running direct in the accurately ground bore of the primary shaft. Two double-purpose bearings, which deal with both journal and axial loads, support the layshaft. Fig. 5 shows a model 437 gearbox mounted in actual position in a chassis with a Perkins P.4 engine.

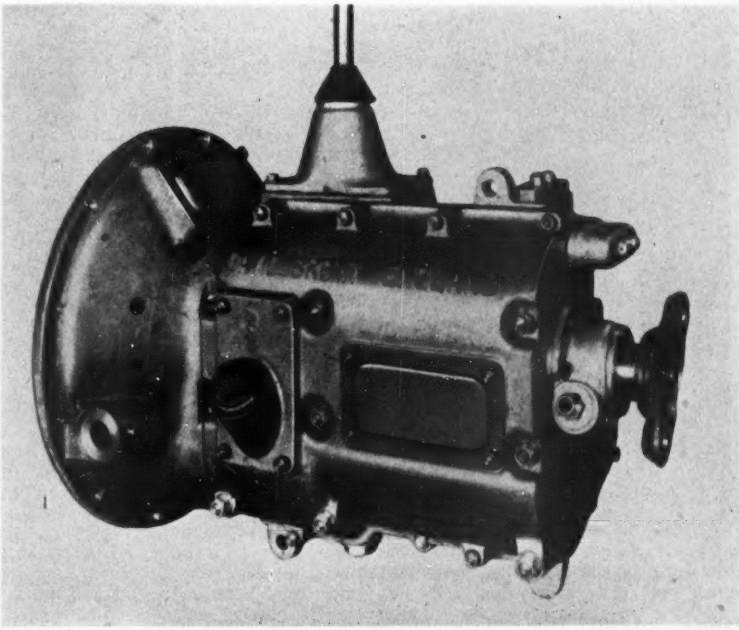


Fig. 4. David Brown model 437 gearbox arranged for central control

Future trend of design

Whilst changes in styling of commercial vehicles do not occur as rapidly as in the case of automobile design, commercial vehicle development is taking place all the time and in line with this, constant efforts are in hand to improve and develop gearboxes. Five speed boxes for medium and heavy vehicles are now very common. Moreover, the present trend is to eliminate all crash gears, making the gearbox fully constant mesh, by the employment of dog clutches, even to the extent of the reverse gear. David Brown and Sons (Huddersfield) Ltd. now have in production a fully constant mesh 557 gearbox. The prototypes have been extensively road tested and the decision to proceed with production of this model was based on the results obtained.

Bad driving in the form of careless engagement of sliding or crash gears can set up very rapid deterioration of the teeth, as this form of gear is very susceptible to chipping at the ends of the teeth, with consequent rapid breakdown of the case-hardened surface. Tempering the ends of the teeth is a partial solution to this problem, but the most satisfactory answer is the constant mesh type box.

Present indications point to the increasing application of synchromesh boxes. It is of course said that these are too expensive; that they will be more liable to go wrong owing to the bigger number of more complicated working parts; and that they are simply a device to make gear-changing more simple for the bad drivers who will not or cannot change gear properly on crash boxes.

Against this, whilst it is agreed that the first cost is heavier, development of synchromesh boxes for commercial vehicles has reached the stage where, providing the synchromesh mechanism is made sufficiently robust, reliability can be expected. A David Brown synchromesh gearbox examined recently had completed 128,800 miles without trouble or servicing of any kind and showed no sign of possible failure of the synchromesh mechanism, whilst the clutch teeth were in good condition and gave no indication of any chipping. The synchromesh mechanism prevents damage to the gears through bad gear changing—a very important consideration in view of the considerable number of crash box failures attributed to bad gear changing. At the moment the demand for synchromesh gearboxes is much more pronounced among builders of the lighter type of commercial vehicle. Synchromesh gearboxes are already, of course, extensively used on the bigger omnibus chassis and there is also a trend in this direction on the lighter omnibus.

Another possible future development on the larger type of vehicle is the use of multi-speed boxes giving from seven to ten speeds. This increase in the number of speeds can also be accomplished by the use of an auxiliary two-speed gearbox in conjunction with a four or five-speed box. The two-

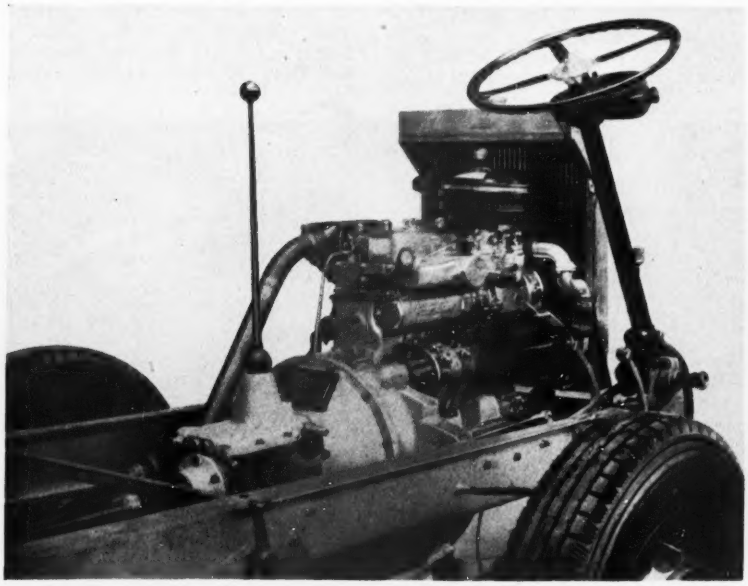


Fig. 5. Typical mounting of a 437 gearbox with a Perkins P.4 engine

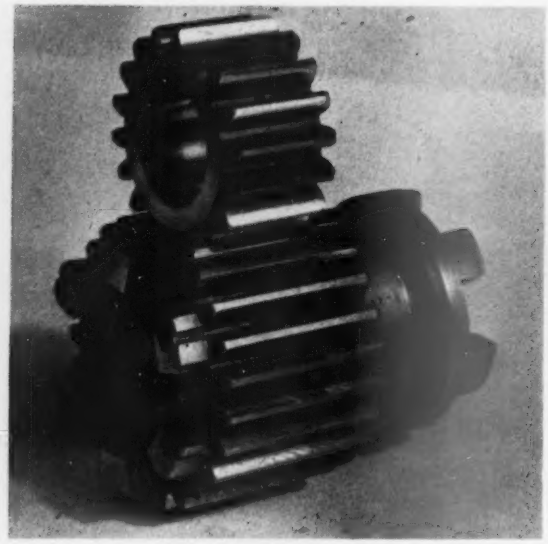


Fig. 6. David Brown gears after 128,000 miles service

speed auxiliary box can be placed either in front of or behind the main gearbox. When placed in front, the auxiliary box is small owing to the smaller torques which are involved in comparison with one placed behind the main gearbox or in the axle.

The David Brown synchromesh gearbox referred to earlier was designed with an epicyclic two-speed portion in front of the main part of the gearbox, making the complete box a multi-speed unit. Fig. 6 illustrates two of the gears taken from the front epicyclic part of the box after 128,000 miles running. It shows the good condition and finish of the teeth even after this length of service. Multi-speed boxes are more widely used in the U.S.A. They are particularly advantageous on the long runs experienced in that country where a return journey may have to be made with the vehicle lightly loaded, a condition suiting the use of the over-drive portion of the gearbox.

At the 1952 Commercial Vehicle Show at Earls Court no vehicles were shown fitted with automatic transmission, and in spite of the extensive use of this type of gearbox in the U.S.A., particularly on passenger coaches, there are no signs of its use in the near future in this country on commercial vehicles. The huge sums necessary for the setting up of the necessary production plant (a sum of £2,000,000 has been mentioned), together with the increased fuel costs (due to the inherent characteristics of this form of transmission, particularly where a torque converter is involved) seem to preclude the immediate possibility of its development here, where low operating costs, covering running replacements and servicing, are a primary consideration among purchasers of commercial vehicles.

Manufacturing operations

In general the machines are grouped for line production. The first operation on the cases, which are generally in halves, is to machine the joints, and Fig. 7 shows a Kendall and Gent vertical mill executing this work. Following this the cases are jig drilled on Asquith or Archdale drilling machines with location taken from faces machined during the previous operation. The end faces of the case are next machined on a Sundstrand Duplex miller as shown in Fig. 8. Boring of the cases is carried out on Richards and Kearns boring machines, in fixtures suitable for each type of case. All remaining drilling and tapping is done on single spindle Archdale or Asquith machines. Bell housings and clutch housings are machined on vertical turret lathes as shown in Fig. 9. The engine facing register diameter on these two items is initially roughed out, leaving 0·025 in for finishing to correct size after all other machining operations are completed. This obviates the distortion which was found to arise when the register diameter was finished in one operation. Alternatively, in some instances, the register could be finish machined in position with the case.

Gear blanks, which are supplied to the machines in the form of upset drop stampings and shafts, are first given a normalizing and annealing treatment as an insurance against distortion which may take place later during the case hardening process. The blanks are then turned on Herbert automatic chucking lathes and Lodge and Shipley Duomatic machines respectively. Fig. 10 shows a No. 4 Herbert automatic chucking lathe tooled for turning a gear blank. It is very important that the boss faces on a gear blank are exactly at right angles with the outside diameter of the blank and parallel with each other, as these faces are often used as locating faces when cutting the teeth. To ensure this requirement an extra light finishing cut is taken on these faces on a Duomatic with the blank mounted on a mandrel between centres.

Wherever possible the teeth in the gears are hobbed. Where the clearance on a gear will not permit run-out for a hob, the teeth are cut on gear shaping machines. A small allowance is left on the flanks of the teeth to enable shaving to follow the gear cutting operation. This allowance is of the nature of 0·0025 in on each flank and should be just sufficient to enable the shaving

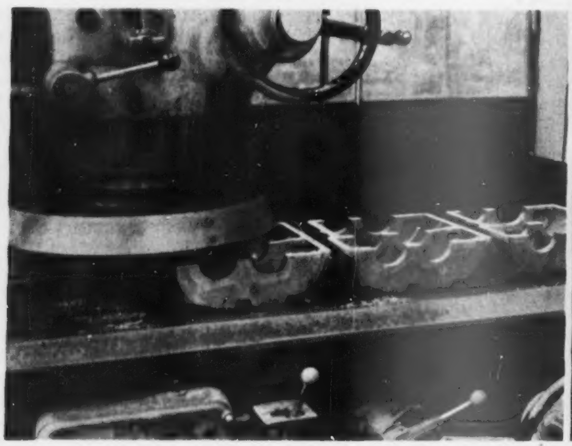


Fig. 7. Machining gearbox joint faces on a Kendall and Gent vertical mill

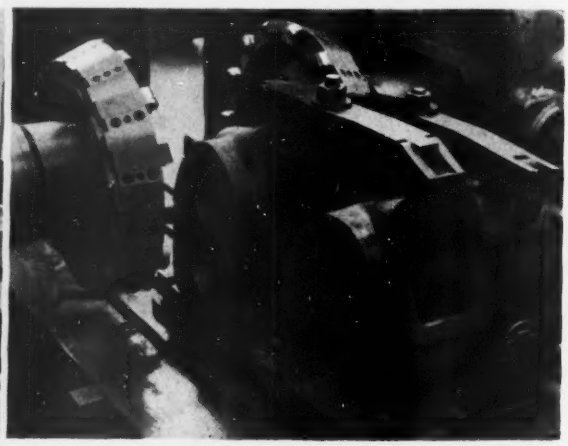


Fig. 8. Milling the end faces of the casing on a Sundstrand Duplex machine

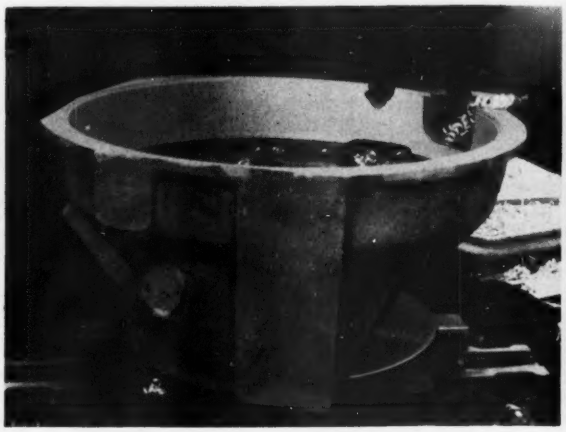


Fig. 9. Machining the bell housing on a vertical turret lathe

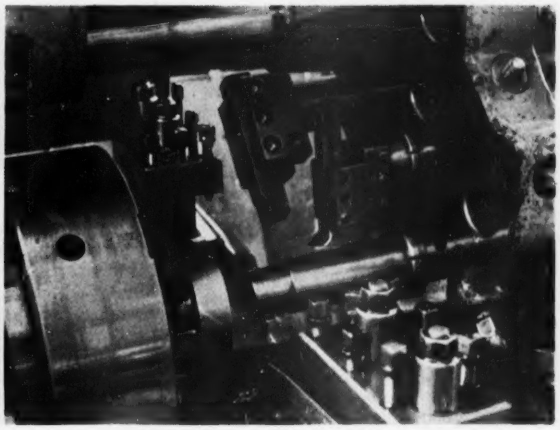


Fig. 10. Tooling for machining a gear blank on a Herbert automatic chucking lathe

cutter to clean up the surface left following the tooth cutting operation. Wherever possible all gears are shaved, thus removing the small errors left by the tooth cutting operation, resulting in quieter gears with longer life. This operation is regarded as most important since, in addition to shaving the teeth, it is possible also to relieve them at the ends at the same time. The relieving effect, which amounts to removing approximately 0·0002 in from each flank at the ends of the teeth, obviates localized heavy contact which could be caused by distortion due to hardening or misalignment of case bores. Shaving eliminates the grinding of the flanks of the teeth after hardening, thereby reducing production costs, whilst in addition a shaved hardened gear is suitable for dealing with somewhat heavier loads than a hardened gear ground on the flanks.

Following case hardening and shot blasting, the gears are ground in the bore and it is very important that the bore runs true with the teeth. This is accomplished by mounting the gears in pitch line chucks. Alternatively the outside diameter of the gear should be ground true with the teeth, then the bore can be ground using the outside diameter as the location.

Although every care is taken in moving gears along the production lines, a proportion inevitably receives minor damage to the teeth. Special precautions are taken to see that these are removed before the gears are passed forward for assembly into gearboxes. All damaged gears have the flanks of the teeth stoned by hand. These gears are then tested with a master wheel on a Parkinson testing machine to check that all damage has been adjusted, and that the gears are correct for concentricity and backlash.

Assembly of gearboxes is done in batches, and the necessary parts to cover the desired production programme are laid out by the stores personnel in convenient positions. Assembly of the gearboxes in the biggest demand is broken down into eight separate sub-assemblies, which experience has shown to be the cheapest method of production.

Each box is then given a light load running test, operating in each speed for a set period, thereby checking that the change speed mechanism is functioning correctly and that the noise produced is within the acceptable limits. Any previous undetected eccentricities in the mounting of the gears and shafts in the casing, or damage to the teeth will soon become apparent during this running test. Discovery at this late stage is obviously an expensive matter, as it may necessitate complete dismantling of the gearbox.

The machines are grouped so that inspection takes place at fixed points, whilst in addition a patrol inspection is employed on each section. These patrolling inspectors select at random any part in production on the section and make an immediate check of the particular operation in question. All drawings state that the first item or operation completed on any batch should be inspected and passed before proceeding with the bulk quantity. The vital importance of this precaution is obvious. Furthermore, the inspector forms a very important link in the premium bonus system on which all machine and assembly personnel operate, as it is only against the inspector's signature indicating the number of correct pieces that bonus payments are made.

Inspection of material received at the inwards receiving department is another important function of the inspection department and it is very necessary that such items as case castings, which are purchased from outside suppliers, should be correct before passing through the machine shop, as any casting errors not revealed until machining is actually taking place can prove very costly.

On completion of inspection all finished items are sent to the stores ready for issuing to the assembly department in the required batch quantities.

All wearing parts in a gearbox necessitate the supply of spares and the production schedule should allow for this. Past experience is a very good guide to the production control department as to quantities and items likely to be required in this respect.

In order to maintain both quantity and quality production it is very important that periodic inspection checks are made on each machine to ensure its accuracy is being maintained whilst, in addition, all jigs, tools and gauges should also be the subject of regular inspection checks. The use of each machine is recorded weekly to enable the supervisor to assess his surplus capacity and these figures, together with the previously mentioned inspection reports, present essential information when a decision has to be taken regarding the replacement of machines.

Service

There are several major troubles which may arise under working conditions unless precautions are taken by the user to see that the gearbox is functioning under proper working conditions.

1. *Oil leakage.* This is often caused by overfilling the gearbox and it is very necessary to adhere to the oil level marking on the dipstick. In some cases the filler plug is placed on the side of the case and the box should be filled up to the level of the filling aperture and topped up to this level at each examination. After the initial filling the box should be drained and refilled with fresh oil after the first 1,000 miles. The oil should then be topped up every 1,000 miles and periodically changed every 20,000 miles. A magnetic oil filter is fitted to the oil drain plug and this should be washed in paraffin before replacing after each oil change. Care should be taken, when covers and companion flanges are removed and re-assembled, not to damage oil seals as the smallest cut

or bruise on a seal can cause oil leakage.

2. *Jumping out of mesh of gears.* It is very essential that the engine and bell housing facings and registers are assembled square and concentric with the crankshaft. This is an important factor in preventing gears from coming out of mesh in service. One has heard of engine facings up to 0·030 in out of square. Excessive shaft deflection under maximum load conditions is also a cause of jumping out of mesh, and to reduce this shaft deflection to a minimum, boxes with centre bearings on both the mainshaft and layshaft were introduced.

3. *Noise.* The volume of noise in commercial goods-carrying vehicle gearboxes is not generally considered so vitally important as in the case of automobiles but in passenger coaches and buses it must be kept to a minimum. Gearboxes are often wrongly accused of being noisy, when the real trouble is external and in some cases may be traced to critical engine speeds. The errors in alignment and concentricity mentioned previously under the last heading can also be the cause of excessive noise in service.

4. *Chipping of gear teeth.* Some drivers experience difficulty in making a clean gear change on crash gearboxes, especially those with sliding gears. Constant crashing of the ends of the gear teeth is bound to bring serious after-effects, as the particles of metal dislodged from the teeth are likely to cause further damage to the remaining gears and bearings even though a magnetic filter is fitted, as this will only collect a proportion of the metal particles. The entry ends of both the sliding and fixed gears are damaged with bad gear changing and if serious chipping occurs the length of gear contact is reduced to a point where failure will occur. Dog engaged gears giving constant mesh of the gear teeth are far superior to sliding gears, as damage due to crash engagement does not affect the teeth. Also, because of the slower peripheral speed of the dogs compared with the peripheral speed of the teeth, the dogs are not likely to be damaged to the same extent as the ends of the gear teeth, if sliding gears were used.

5. *Other possible gear tooth failures.* When gearboxes commence their working life the surfaces of the gear teeth contain high spots which are smoothed down in service by the local breakdown of the oil film. This is due to the high local working pressures which arise at these points, and under normal conditions further abnormal wear does not arise. The use of an excessively light oil could, however, be the cause of further rapid wear, and this may so reduce the thickness of the teeth that further use is impossible. Breakdown of the oil film may also induce localized welding of gear teeth, causing metal to be lifted from the flanks of the teeth. A heavier oil should be tried under these circumstances. A straightforward fracture of a gear tooth often indicates the sudden application of an excessive load, such as allowing the clutch to engage with a jerk.

OVERHEAD VALVE GEAR

IN a paper entitled "Overhead Valve Gear Problems" by R. P. Horan, in an *S.A.E. Preprint*, March 3-5, 1953, the author studies valve dynamics by considering the valve train as a vibratory system with a single degree of freedom as shown in a figure, which also gives the lift and acceleration curves of a typical cam. Vibratory conditions occurring as the valve closes are examined by an analysis of the shape of the appropriate portion of the acceleration curve. Critical cam speed is found to depend upon the length of the acceleration cycle and the natural vibratory frequency of the valve gear. To keep the vibration frequency high, the stiffness-mass ratio of the valve gear must be large. Increases in the spring load with the valve closed and the use of spring dampers of various types permit satisfactory engine operation at higher speeds. The spring load, however, cannot be increased indiscriminately because of tappet face wear considerations.

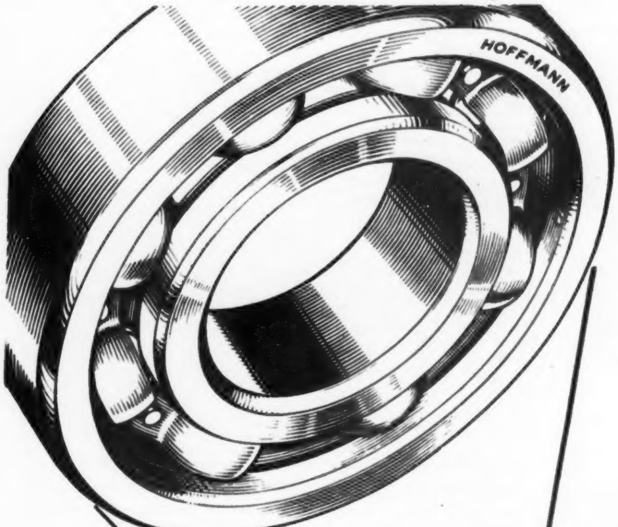
A good valve gear design, in addition to giving the desired dynamic characteristics, must obtain as high a valve lift as possible with a minimum restriction on flow. A review of methods of achieving these conditions is given, together with a discussion on the question of valve overlap. The advantage, under certain conditions, of using individual exhaust manifold branches for adjacent cylinders is also explained. From a chart based on a considerable amount of engine data, it is possible to determine whether the performance of an engine accords with that to be expected from the induction and exhaust port diameters.

Variables influencing tappet face failures are: face stresses, alignment, material combinations, finish and manufacturing techniques, surface coatings and lubricants. A figure shows a considerable stress range for both overhead valve and L-head engines in comparison with a test fixture in which all other variables were carefully controlled. For overcoming the problem of tappet alignment, a "hyperbolic" tappet having the side contoured to permit the tappet face to accommodate itself to any misalignment has given encouraging laboratory results. Among material combinations, a chilled cast iron tappet face run against a steel camshaft gives the best results.
(*M.I.R.A. Abstract No.* 6278.)

NEW JOINT WASHER

IN the June 1953 issue of *The Engineers' Digest*, a new type of joint washer, termed the "Flow-in" joint washer, is described. It has recently been introduced as a replacement for conventional pre-cut cork or rubber washers, and is said to provide an improved seal at lower labour and material cost. It is applied as a liquid synthetic rubber or resin compound forced through a nozzle on to a spinning component part and then baked to form an elastic washer that will not fall off. The washer can be applied either automatically or semi-automatically, depending upon the lining equipment used, and in varying types, thicknesses and diameters. The manufacturers claim that the chief advantages over conventional pre-cut washers are lower material costs and faster application since the washer does not have to be crimped, stamped or glued to hold it in place.

On the automatic lining machines, with which washers can be applied at a rate of as much as 300 per minute, depending upon their size, the components are belt or gravity fed to the machine. They are then passed one at a time to the chuck and are rotated under an adjustable nozzle that squirts a measured amount of compound into a groove on the joint face. The lined component is then moved off on a conveyor belt to an oven where it is baked. The machine can handle components from 3½ in to 12 in diameter. If a belt-feed is used, high speed equipment is employed and no operator is required, but when a semi-automatic machine is used the component is placed in the chuck by hand. An advantage of the new method appears to be that specially formulated compounds may be used to give effective seals against moisture, oils, petrol, vacuum, vibration, etc., as the case may be. (2049)



Where ACCURACY is essential

HOFFMANN

BALL AND ROLLER
BEARINGS
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For every component from gear box to universal joint, accuracy is very essential. Smooth running and reliable performance are also ensured by the use of Hoffmann products

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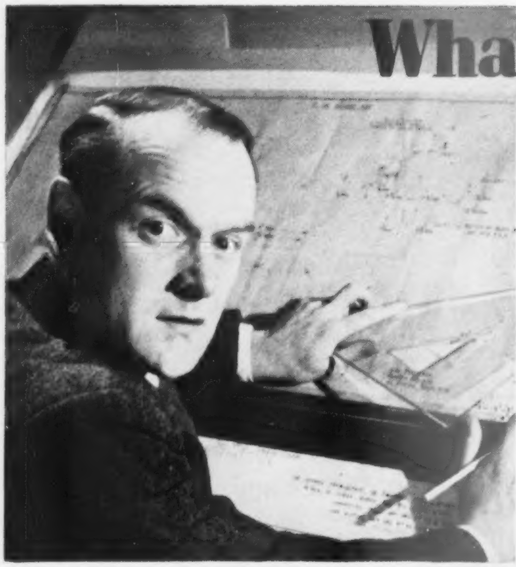

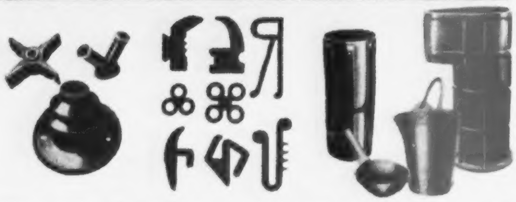

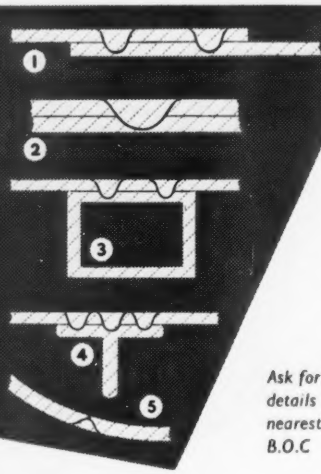

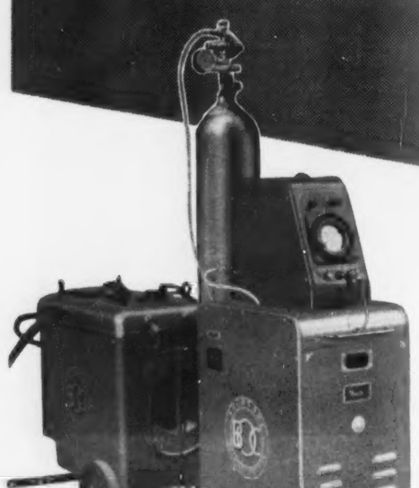

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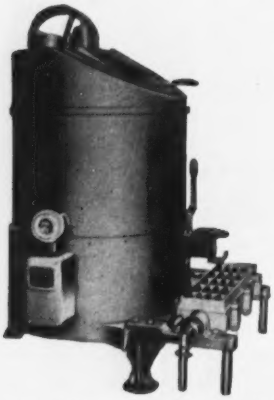

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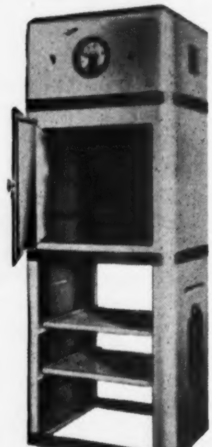

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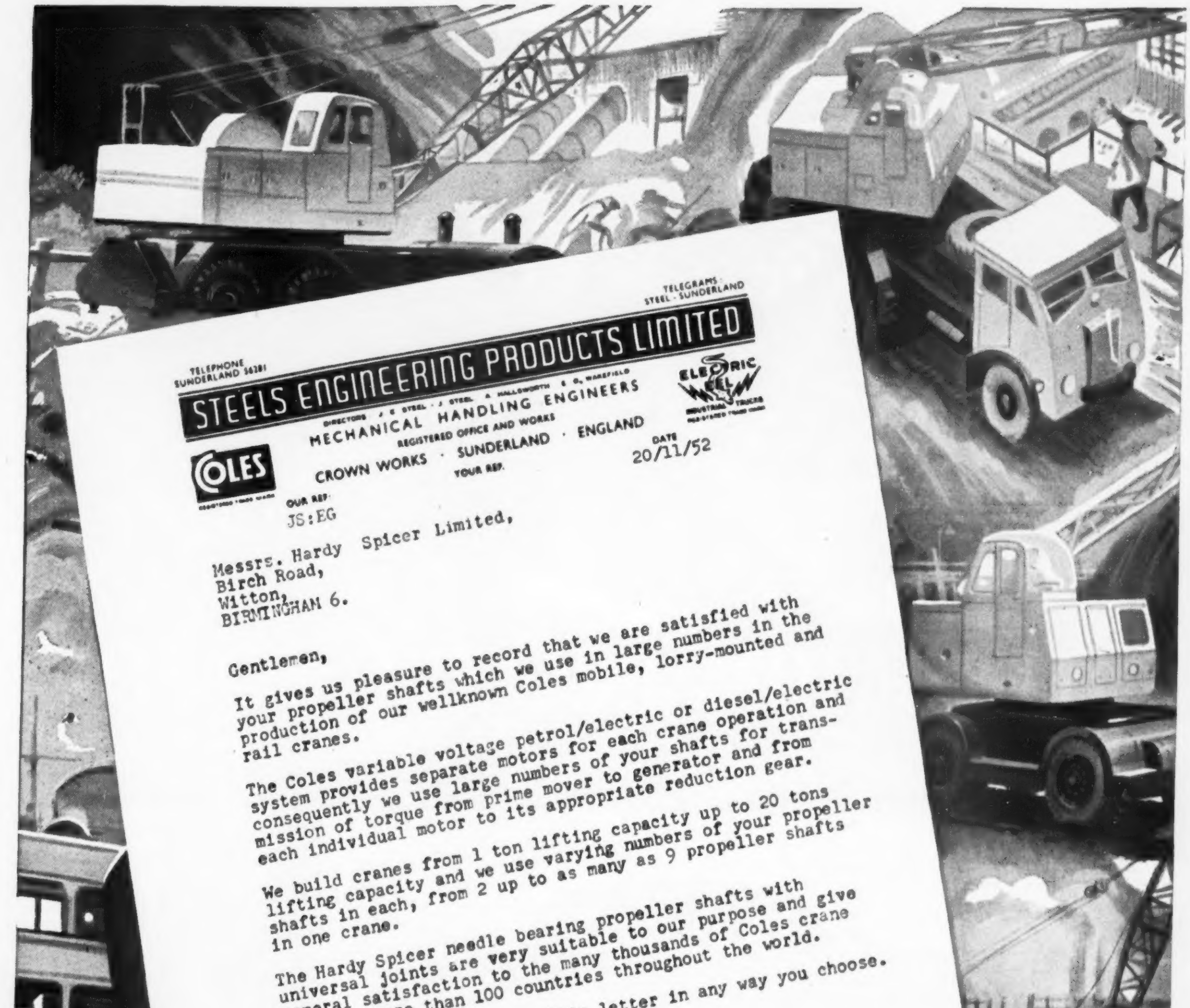

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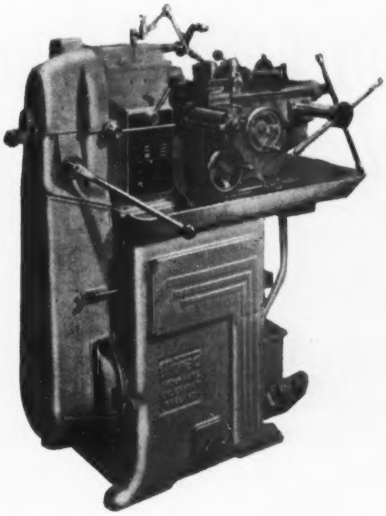

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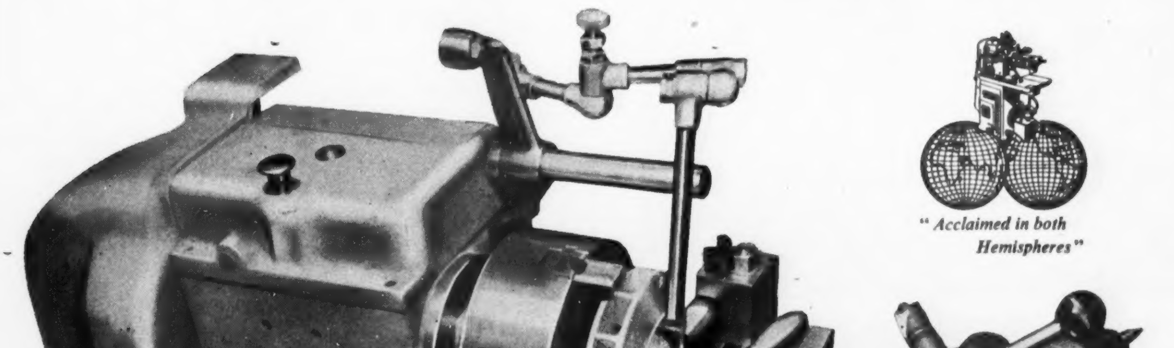

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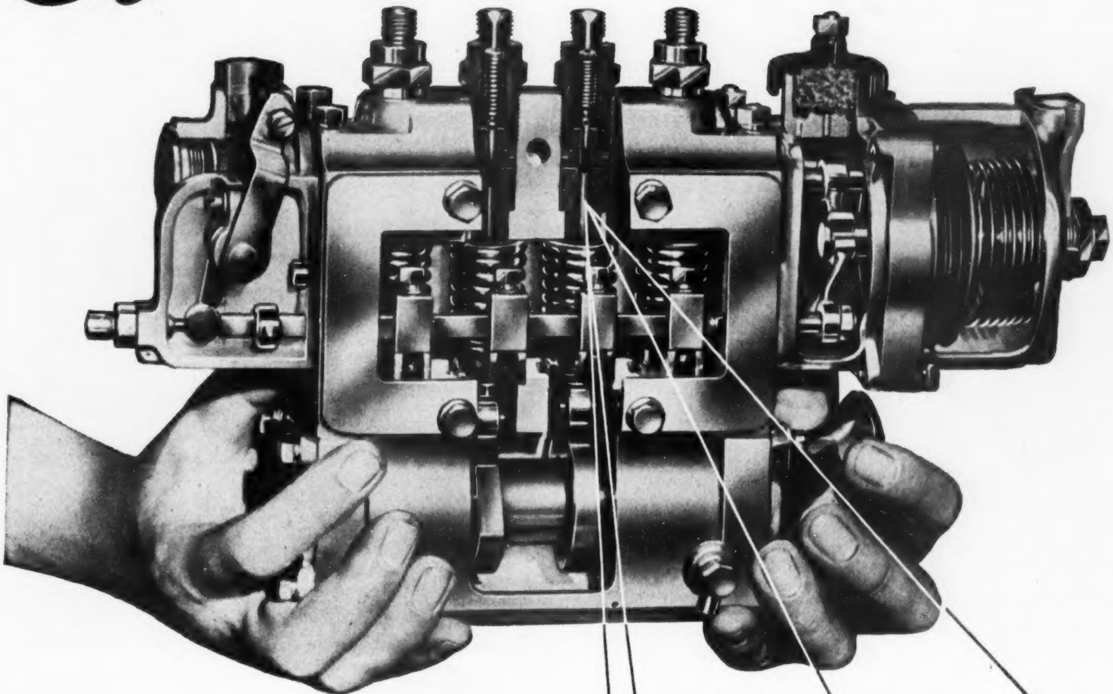
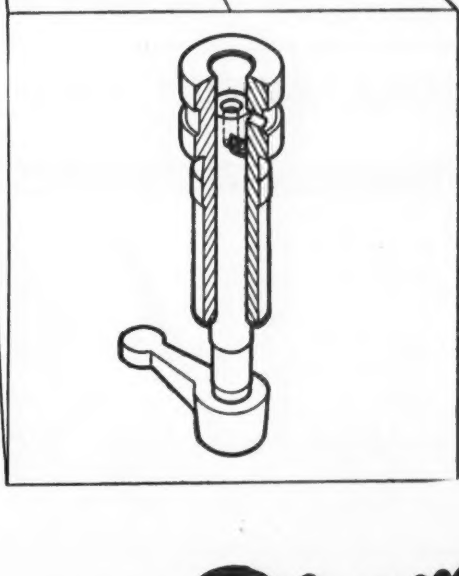
Simms *present* the new features of their fuel pumps in detail





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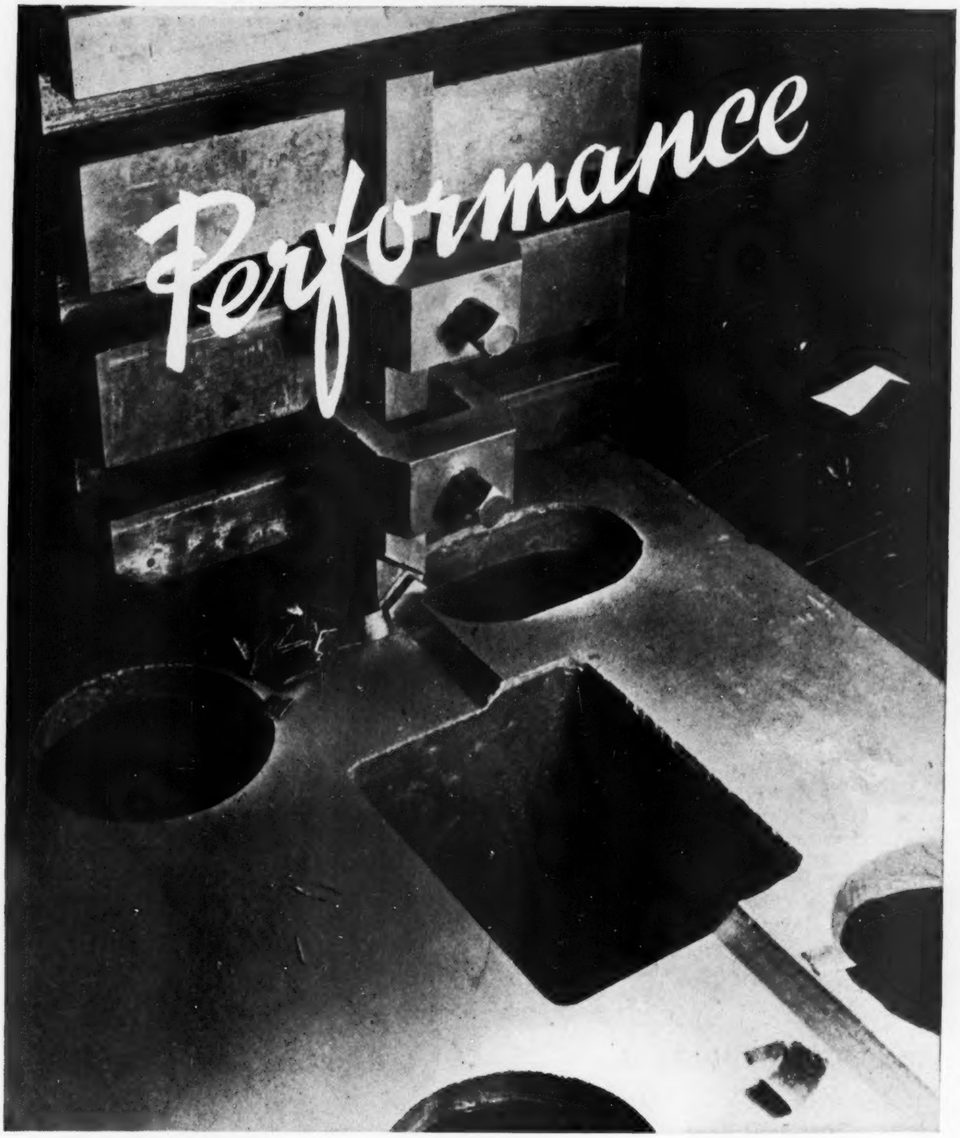

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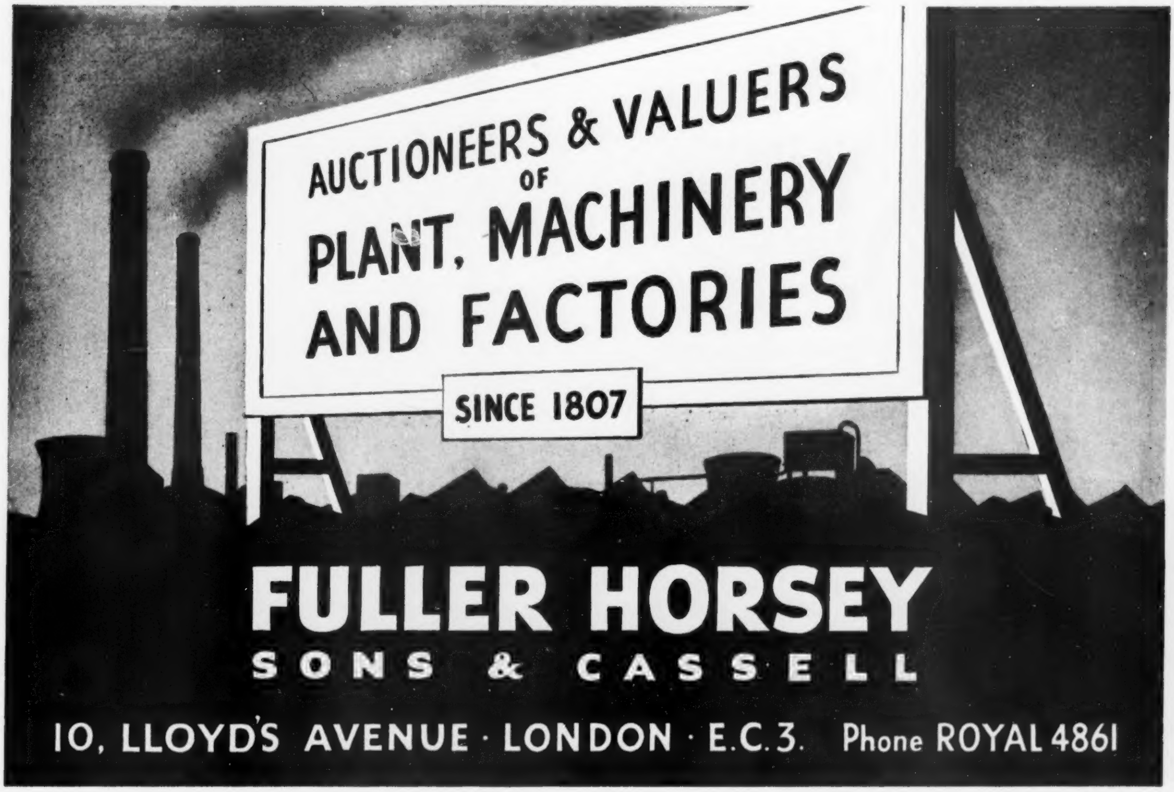

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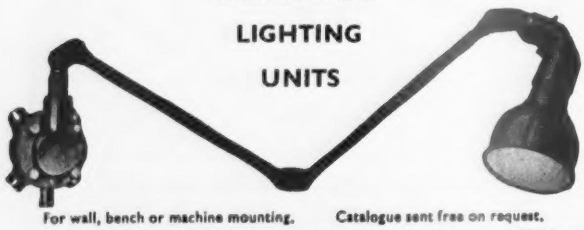

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The engagement of persons answering these advertisements must be made through the local office of the Ministry of Labour and National Service etc. if the applicant is a man aged 18-64 or a woman aged 18-59 inclusive, unless he or she or the employer is excepted from the provisions of The Notification of Vacancies Order, 1952

SITUATIONS VACANT

SENIOR Designer required for Structural or Airframe Section of Guided Weapon Design Office. This man will be required to work on the Production Design of an accepted principle and applicants with both aircraft and/or motor industry experience will be considered most suitable. Ordinary or Higher National Certificate desirable. This post offers scope for advancement in an expanding organization. Address replies to Dept. C.P.S., The English Electric Co. Ltd., 336-7, Strand, W.C.2, quoting reference 845A. [4690

DESIGNER Draughtsman required for research and development department of large Midland motor firm. Write stating training, experience and qualifications. Box 8804 c/o Automobile Engineer. [4703

MINISTRY OF TRANSPORT: CERTIFYING OFFICER. The Civil Service Commissioners invite applications for 4 pensionable posts. Vacancies are in London, Birmingham, Leeds, Manchester. Age at least 35 on 1st January, 1953. Candidates must be Corporate Members of the Institution of Mechanical Engineers and must be fully qualified mechanical and automobile engineers who have completed a recognised professional training.

INCLUSIVE salary £970 to £1,280. Somewhat lower for women and in provinces. Promotion prospects.

FURTHER particulars and application forms from Civil Service Commission, Scientific Branch, Trinidad House, Old Burlington Street, London, W.1, quoting S 4230/53. Application forms to be returned by 27th August 1953. [4704

ENGINEER Designer required by old-established firm, aged 30-40, with some scientific training; University graduate preferred. Applicants must have had considerable experience in the design and development of small diesel marine propulsion engines up to 300 b.h.p. Workshop experience essential, and the candidate should be a capable draughtsman able to prepare designs, carry out all necessary calculations and be familiar with up-to-date testing methods and general experimental practice. This is a permanent post with good salary, pension scheme, and excellent prospects. Replies should give brief history of training and experience. Box 8808 c/o Automobile Engineer. [4716

AN experienced man is required to manage engine test dept. He should be capable of drawing up and carrying out test programmes, including instrumentation, etc. Apply in writing to The Chief Engineer, E.R.A. Ltd., London Road, Dunstable, Beds. [4698

TECHNICAL Executive. A vacancy occurs on the staff of a London manufacturer of specialised light electrical equipment for automotive and similar applications. This is a senior engineering appointment with opportunities for advancement, carrying responsibility for the direction of development through all stages to final quantity production. Applicants should be qualified engineers preferably under 40, who can show evidence of real administrative and commercial experience. In reply quote TE/1. Box 8806 c/o Automobile Engineer. [4714

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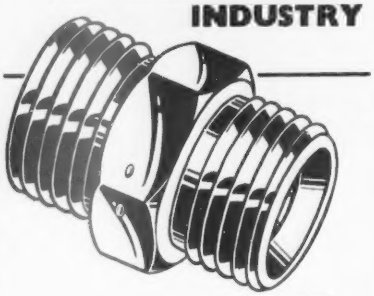
ANGLAMOL Technical Service Laboratories, Hazelwood, nr. Derby, specialising in lubricating problems and oil development have a vacancy for a Technical Assistant. Preferably Degree standard and with good knowledge and enthusiasm for internal combustion engines, must be capable of writing a good report. This vacancy represents an excellent opportunity for the right man. Salary in accordance with qualifications and experience. [4713

SENIOR Designer required with experience of heat exchange problems. Capable of designing radiators for heating or cooling installations for industrial purposes or vehicles, and able to control a small section of Drawing Office on such problems. Write giving particulars of age, experience and salary required to: The Chief Engineer, Clayton Dewandre Co. Ltd., Titanic Works, Lincoln. [4712

SENIOR Draughtsman required with experience in the design and manufacture of pneumatic and hydraulic brake components. Write giving particulars of age, experience and salary required to: The Chief Engineer, Clayton Dewandre Co. Ltd., Titanic Works, Lincoln. [4711

PRODUCTION Manager. An old-established Midland company wishes to appoint a really experienced Engineer to take full control of all production departments in connection with the introduction of a new high grade sports car. The position offers great scope to a first-class man having adequate experience of both chassis and body engineering, and he will be required to engage his own team for the purpose. Applications will be treated in the strictest confidence and should be addressed to "The General Manager," giving full details of education, career, qualifications and salary required. Age group 35-50 years. Box 8805 c/o Automobile Engineer. [4707

SHEFFIELD area. Designer-Draughtsman, age 30-35 years, for automotive work, preferably with experience in power-transmission layout for road and rail vehicles. Box 8807 c/o Automobile Engineer. [4715



SITUATIONS VACANT

AUTOMOBILE Service Engineers. Technical Assistants, Claims Correspondent and Instructors required. See Ford Motor Company Ltd.'s displayed advertisement in this issue. [4710

CHASSIS Draughtsman required by the Rootes Group Truck Plant in Luton. Applicants must have a wide experience of medium and light vehicle chassis design.

PERMANENT position with good salary and pension scheme.

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SENIOR Planning Engineer and Draughtsman required for all metal coachwork on Rolls-Royce and Bentley chassis. Write stating qualifications etc., to Managing Director, Jack Barclay Ltd., Berkeley Square, London, W.1. [4705

AUTOMOBILE Engineer. Joseph Lucas Ltd. require a first class engineer to take charge of the section of their laboratories dealing with investigations into the starting of I.C. engines under low temperature conditions and allied problems. This position provides an excellent opportunity for a young energetic engineer with previous engine testing experience, preferably including cold room work. Applicants should have an engineering degree and should be not older than 35 years of age. Please reply stating qualifications, experience and age to Personnel Manager, Joseph Lucas Ltd., Great King Street, Birmingham, quoting reference PM/D/75. [4702

OLD Established Midland Company require a fully experienced automobile designer competent to take charge of small drawing office under the direction of the Technical Manager.

PLEASE give summary of experience and salary required, addressed: For the attention of the Technical Manager, Box 8802 c/o Automobile Engineer. [4700

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AGENCY/DISTRIBUTORSHIP, Automobile components or accessories, required by firm already wholesale distributors for well-known accumulators in area Bournemouth, Christchurch, Poole, Weymouth, Dorchester and Salisbury. Box 8803 c/o Automobile Engineer. [4700

PATENTS

THE proprietor of British Patent No. 601749, entitled "Improvements in or relating to air cleaners," offers same for licence or otherwise to ensure practical working in Great Britain. Enquiries to Singer, Stern & Carlberg, 14, East Jackson Boulevard, Chicago 4, Illinois, U.S.A. [4699


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